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IRRADIATED FUEL TRANSPORTATION SHOCK
AND VIBRATION PROGRAM-PHASE II SUMMARY

Report No 85-190-K

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ABSTRACT

The Irradiated Fuel Transportation Shock and Vibration Program is part of Ontario Hydro's overall Fuel Management Transportation Program under the direction of the Design and Development Division. The ultimate goal of the Transportation Program is to specify off-site transportation systems suitable for unrestricted transport of irradiated fuel by road, rail and water. This report summarizes the experimental and analytical components of the Transportation Shock and Vibration Program that were necessary to reach the objectives of the second phase. An assessment of the adequacy of transportation system performance with respect to shock and vibration is now possible.

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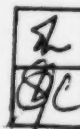
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EXECUTIVE SUMMARY

Irradiated fuel bundles are an accumulating by-product of Ontario Hydro's CANDU nuclear generating stations. The current plan is to store the irradiated fuel at the station until a national decision on reprocessing for plutonium extraction or long-term disposal is made. This is expected to occur after the year 2000. In the interim, however, an operational off-site transportation system to demonstrate the feasibility of transporting significant quantities of CANDU fuel will be put in service by the late 1980's.

In order to specify off-site transportation systems suitable for unrestricted transport of irradiated fuel by road, rail and water, Design and Development Division initiated the Irradiated Fuel Management Transportation Program in 1975. The Transportation Shock and Vibration Program is to provide assurances that fuel bundle and shipping module integrity are maintained during normal transport elsewhere. By combining the results of the experimental and analytical program components described in this report, a positive assessment of these objectives was made.

As a result of this phase of the Shock and Vibration Program, physical reference data as well as analytical models of transportation system configurations have been produced. This information will be used for the design and development of transport systems optimized for performance and economy.



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To Mr. H.N. Isaac
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IRRADIATED FUEL TRANSPORTATION SHOCK AND
VIBRATION PROGRAM - PHASE II SUMMARY

1.0 BACKGROUND

Ontario Hydro is presently developing a transportation system for the safe transfer of irradiated fuel on a production basis from its Candu nuclear power stations to a suitable disposal site. By late 1985, the basic design specifications are to be completed with a prototype road cask to be designed, built and licensed by 1988. The elements of the transportation system as defined for this program are the vehicle, the cask, the shipping and storage module (which includes the fuel) and the cask tie-down.

The overall Transport Cask Program is under the direction and responsibility of the Nuclear Materials Management Department of the Design & Development Division. The Irradiated Fuel Transportation Shock and Vibration Program forms a part of the overall Program and addresses the requirements of a transportation system related to the normal handling and shipping of irradiated fuel. Considerable work done during "Phase I" of the Program is summarized in Reference/1/.

The present report summarizes the second phase of the Shock and Vibration Program in which more detailed tests and analyses provide the required input for the design of a transportation system. This report constitutes Activity List Task 14 of the Transport Cask Program and contains details of Tasks 4 and 8. It also summarizes Reference/2/ which details Tasks 11 and 12. In addition, this report summarizes a series of static load tests on shipping modules to verify their structural integrity during relative fuel bundle motion within the modules under transportation loads. (This work was not itemized in the Transport Cask Program List.)

2.0 PHASE II SHOCK AND VIBRATION PROGRAM

The Phase II Shock and Vibration Program is essentially an extension of the Phase I Program with requirements for specific details related to the design of a transportation system. In order to greatly simplify the design of the cask internals, for example, a proposal was made to provide no axial preload or restraints (other than the cask walls themselves) on the fuel bundles. This required that impact and fatigue thresholds for irradiated fuel bundles be re-established under the appropriate test conditions. With the bundles free to move axially (to a limited degree), an axial loading on each module tube due to sliding friction was possible. A re-assessment of the structural integrity of the shipping module was thus necessary. These and other requirements of the Phase II Program are outlined in Reference/3/.

The results of the Phase II work were initially transmitted via memorandum in June 1983/4/. The basic conclusions reached remain unchanged.

2.1 Objectives

The objective of the overall Shock and Vibration Program is to ensure that the irradiated fuel bundles and the shipping and storage modules will not fail due to shock and vibration during normal off-site transportation by road, rail or water. "Failure" in this Program is defined as any distortion or disassembly of fuel bundles or modules which will prevent automated handling at the disposal site. Local bundle defects such as endplate or sheath cracks do not in themselves constitute failure, provided automated handling is not affected.

The individual objective for each of the Tasks in the Phase II Program are listed below:

- (i) Task 4 - Low Temperature Brittle Fracture of Fuel: The objective of this Task is to determine if there is any detrimental effect on the impact or fatigue strength of fuel bundles when transported at -30 C. This is the minimum anticipated fuel temperature under the worst conditions, ie winter travel, old fuel, and high vehicle speed.
- (ii) Task 8 - Fuel Ability To Survive Normal Shock and Vibration (with axially unrestrained bundles): The objective of this Task is to determine first the threshold strength of irradiated fuel bundles in impact and fatigue under conditions of axial clearance of up to 33 mm (15/16 inches). These threshold levels are then compared to estimates of the levels encountered during normal transportation to determine fuel survivability (the question of fuel survivability is scrutinized in more detail under Task 14).

- (iii) Task 11 - Shock and Vibration Input from Vehicle/Tiedown: The objective of this Task is to determine the normal off-site transportation shock and vibration input to the module from the vehicle/tiedown configuration.
- (iv) Task 12 - Shock and Vibration Transmission Through Module Restraint: The objective of this Task is to determine the normal off-site shock and vibration transmission through the module.
- (v) Additional Task - Axial Loads Imposed on Module Tubes During Normal Transportation: The objective of this Task is to determine the effect of the axial force generated by an axially unrestrained sliding bundle at the bundle-to-tube contact surfaces on module structural integrity. If these sliding forces are significant, tubesheet distortion could occur.
- (vi) Task 14 - Combine Tasks 8, 11 and 12 to Assure Fuel Survives: The objective of this Task is to summarize and compare the findings of all aspects of the analytical and experimental work in order to assure that the fuel survives normal off-site transportation. If there are some conditions in which this is not the case, recommendations to prevent failure of the fuel or module should be made.

2.2 Terms of Reference

The terms of reference of the Shock and Vibration Program have been reported in Reference/3/. They are summarized below and reflect some changes which were necessitated by economic reasons.

(i) Transportation Modes and Vehicles: The three modes of transportation considered are rail, road and water. Since the final storage sites have not been located, mixed-mode transportation may be required to deliver the fuel. For the rail mode, standard flat cars have been specified. For the road mode, standard tractor-semitrailer vehicles are preferred. If fuel survival under normal transportation is not possible with standard vehicles, dedicated vehicles will be examined. Dedicated vehicles may also be considered if accident or high speed impact scenarios dominate vehicle selection.

Transport by water, if eventually required, will be by barge or ship. Because of the low probability of transport by water, and because the shock and vibration environment is expected to be no worse than for the other transport modes, the water transport is not included in this report. It will be investigated using information from an ongoing survey of the technical literature.

ii) Fuel Container (Shipping and Storage Modules): The fuel container concept shown in Figure 1 is used for both on-site storage and off-site transportation of irradiated fuel bundles. Two

bundles are housed in each tube and the total capacity is 96 bundles. The module tube longitudinal axes are aligned in the direction of travel for rail transport. For road transport, the axes are normal to the travel direction. Mechanical design details are available in OH Drawing No 904 DIH 35300 0903.

(iii) Transport Casks: The transport casks will be designed to provide not only radiation containment but also protection of the module/fuel bundle assembly against normal off-site shock and vibration and overheating. The final configuration may be circular or rectangular in horizontal cross section. The proposed concepts are illustrated in Figures 2 and 3. The preference is to have the modules totally unrestrained within the cask, thereby greatly simplifying the internal design of the cask. The internal dimensions are such that the module can be lowered into the cask without any interference between the ends of the tubes and the walls. The bundles will be free to move axially up to a distance of 33 mm (1.31 inches).

The rail cask will weigh about 68 Mg (75 tons), the truck cask about 32 Mg (35 tons). The rail cask may contain four or more shipping and storage modules, while the truck cask will carry a maximum of two.

(iv) Fuel Bundles: Properly hydrided unirradiated fuel bundles have been shown to exhibit fatigue and impact resistance very similar to irradiated fuel bundles/5/. An equivalent Hydrogen concentration of 80 ppm has been established in consultation with OHRD, AECL-CRNL and AECL-WNRE as the appropriate concentration in the endplate and endplate-to-endcap welds to simulate irradiated fuel bundles/6/.

Twenty hydrided fuel bundles were to be used in the test program, but due to budgetary constraints, this number was reduced to fourteen. All test bundles are of the Bruce 37-element type, since these were shown to have a lower fatigue strength than the Pickering 28-element type in the previous hot cell tests/5/.

(v) Fuel Bundle Test Conditions: The same basic test procedure as used in the original hot cell tests (Phase I) on irradiated bundles is used in this test program. The significant differences are:

(a) an axial clearance of 33 mm (1.31 inches) is to be used in all tests, and (b) the axial impact tests are to be done with two bundles to properly account for the interaction between them.

(vi) Temperature Conditions: During winter transportation of old irradiated fuel (low residual heat), the minimum fuel bundle temperature has been estimated to be - 30°C. The two factors which may affect the impact and fatigue strengths at this temperature have been identified as (a) metal embrittlement and (b) differential thermal expansion of the fuel pellets in the diametral direction within the sheath.

3.0 FUEL BUNDLE TEST PROGRAM (TASKS 4 AND 8)

These tasks are reported here in detail as they are not documented elsewhere.

3.1 Task 4 - Low Temperature Brittle Fracture of Fuel

The objective of this test as outlined in paragraph 2.1 was expanded to include any possible effects of high temperature on fuel bundle strength. If the diametral clearance between the fuel pellets and the sheath is increased, the bundle element strength may be reduced due to the reduced stiffness effect of the pellets on the sheath. This may occur at either lower or higher temperatures depending on the relative thermal coefficients of expansion of the two materials. Since these coefficients are approximately $46 \times 10^{-6} / ^\circ\text{C}$. and $6.6 \times 10^{-6} / ^\circ\text{C}$ for natural uranium and zircaloy -4 respectively, a temperature increase would cause an increase in the diametral clearance. Hence tests were made at both lower and higher temperatures than the original fuel bundle impact and fatigue tests.

3.1.1 Literature Survey: A survey of available literature on the variation of fracture toughness of fuel bundle material (Zircaloy -4) containing hydrides with temperature was limited to work currently in progress at OHRD - Metallurgy Department/7/. No other references to similar work have been found.

One of the results of this work is that at Heq levels of 80 to 100 ppm, the range of interest in this program, the relative fracture toughness is at a low level "plateau" at temperatures below about 200°C (see Figure 4). These test specimens, however, had predominantly radially oriented hydrides, ie in the crack direction. The effect of randomly oriented hydrides, as has been shown to be the case in the endplate-to-endcap weld area, might be to lower this brittle transition temperature. The possibility may then exist that the fracture toughness at -30°C is somewhat lower than at the temperature of the fuel bundle tests (about 20°C). There is considerable opinion (reference/8/ and its references) that this effect, if any, is small; nevertheless, no known data at this temperature exists.

3.1.2 Temperature Tests

In order to determine any changes in the fuel element strength due to temperature change, tests were performed on a bundle element over a range of -30°C to $+95^\circ\text{C}$. In this method, any decrease in pellet support stiffness will be detected by a reduction in the natural frequency of vibration of the element.

(1) Procedure: A miniature accelerometer was attached to one of the outer elements of an unirradiated Bruce test bundle at mid-span. The bundle was soaked in a forced draft environmental chamber at each temperature for at least two hours. The element was excited in the transverse direction by an impact with a small

hammer, and the resulting vibration response was recorded and analyzed.

(ii) Test Results: The dominant frequency at the first mode of vibration at each temperature is listed below:

Bundle temperature (°C)	-30	+20	+51	+95
Frequency (Hertz)	36.50	35.38	35.13	36.88

The maximum frequency increase of 4.75% between +51°C and +95 °C represents an increase in stiffness of about 10%. This is not considered to be significant as far as the mechanical strength of the bundles is considered. What is unexpected in these results is that the resonant frequency has a minimum near +50°C and does not vary linearly over the temperature range. This implies that factors other than a change in pellet-to-sheath clearance may exist which affect the stiffness of the bundle element.

3.2 Task 8 - Fuel Ability to Survive Normal Shock and Vibration (Determination of Bundle Thresholds)

A series of impact and fatigue tests similar to those previously performed and reported/5/ were carried out at OHRD on hydrided, unirradiated fuel bundles. A total of 14 bundles were available with six allotted for fatigue tests, six for axial impact tests, and two for lateral impact tests. One bundle, S/N G53667W, was used to establish the hydriding procedure and was metallurgically examined for hydride concentration. Thus, one of the remaining bundles (S/N G53601W) had to be used for two tests. In addition, one pair of bundles (S/N G53601W and S/N G53668) was used for an extra test in the axial impact test series. Table I lists the allocation of each of the test bundles.

The results of the fatigue tests are quoted in terms of root mean square (rms) levels in order to be directly comparable with the results of analytical modelling for random response. Impact results are quoted in peak acceleration units for both tests and analyses.

(a) Fatigue Tests: A total of six bundles were tested in pairs in a test rig which could be adjusted to provide the required axial clearance (33 mm). All other test parameters were the same as in the previous hot-cell tests. The test rig was mounted on an hydraulic shaker to provide vibration input at 21.3 Hz in the vertical direction with the bundles in the horizontal position (see Figure 5). The drive level was monitored by an accelerometer mounted at the top of the apparatus at the mid-point. The bundles were removed for inspection at approximately two hour increments over a total duration of 24 h running time. The "front" bundle is the first one inserted into the rig, the second bundle being the "rear" bundle. The monogrammed ends of each bundle were located at the outboard ends of the test fixture with the monograms at the 12 o'clock (0°) position at the start of each time increment.

The results of the fatigue tests are listed in Table II. Although endplate cracks were formed, all six bundles remained integral and no bundle failures, as defined in the terms of reference, occurred during any of the tests. The endplate cracks all occurred either along the edge of an endplate-to-encap weld heat affected zone or part way between welds. Typical endplate cracks are illustrated in Figure 6. Wear also occurred on the wear pads, especially at the outboard ends of the test bundles. This is illustrated in Figure 7. Figure 7a shows the wear which occurred at the outboard pad of the rear bundle. Because the simulated shipping module tube was not centred between the ends of the fixture, the rear bundle pads partially protruded past the end of the tube, thus causing this wear pattern. Figure 7b shows the more uniform wear which occurred at the opposite end of the front bundle. Note that some amount of wear also occurred at the element end cap.

Because the fatigue threshold for Bruce bundles in the hot cell tests was actually between 3.47 m/s^2 (0.39 g) rms and 5.20 m/s^2 (0.53 g) rms, this fatigue program was started at the lower level. A smaller increment to 4.33 m/s^2 (0.44 g) rms was used to "home in" on the actual threshold. When no failure occurred at this level, it was incremented again to 5.20 m/s^2 (0.53 g) rms. Because of the limited number of bundles available for testing, higher drive levels could not be tested. Hence the threshold level for fatigue based on a 24 h duration for Bruce bundles is at least 5.20 m/s^2 (0.53 g) rms.

The pair of bundles previously tested at 3.47 m/s^2 (0.35 g) rms for 24 h were also tested at a level of 13.9 m/s^2 (1.41 g) rms for a short period of time (0.5 h) in order to approximate the effect of higher level transient inputs which may occur during transportation. Both bundles remained integral and did not even sustain any endplate cracks.

(b) Axial Impact Tests: The test apparatus for the axial impact tests is illustrated in Figure 8, and is considerably different than that used in the hot cell tests/5/. The test fixture contains two bundles which are inserted using the same procedure as for the fatigue tests. The rear bundle is rotated with respect to the front bundle to produce the worst-case alignment of outer elements. In order to produce the appropriate time history characteristics, a four kilogram mass of "Duxseal", a viscoplastic material, was kneaded into a cylindrical shape about 10 cm long. After each impact, the material was re-formed to the original shape and length.

Three clearance modes were tested, and are identified in Figure 9. Mode 1 has all the clearance in between the two bundles. Mode 2 has all the clearance at the front of the bundle pair. Mode 3 has half the clearance at the front and half the clearance in between the two bundles. An extra test was also performed on a bundle pair which had been previously subjected to a fatigue test for 24h at 7.35 m/s^2 (0.75 g) peak. This was done in order to determine if the combination of tests would induce a bundle failure.

The results of the axial impact tests are listed in Table III. Although cracks occurred in some endplates, all bundles remained integral and no failure occurred, even for the two bundles that had been previously fatigue tested. A typical time history is shown in Figure 10. As can be noted from the graph, the initial deceleration peak is followed by a secondary peak of the opposite sign. This is due to the internal impact (or impacts) which occur when the bundles take up the axial clearance in the test fixture. The table lists the rise time from initiation to the time at which the peak deceleration occurs, and also the time from initiation to the time at which the secondary peak occurs. This characteristic is quite different from the single impact events of the hot cell tests, as expected.

It should be noted that the secondary impact was always substantially greater than the initial impact. The maximum values obtained for initial and secondary peaks were -390 m/s^2 (-39.8 g) peak and $+1285 \text{ m/s}^2$ ($+131.0 \text{ g}$) peak respectively, and occurred on the same test (Clearance mode 1, test 8). However, the most damage occurred during the mode 2 tests, when a total of 8 endplate cracks were produced.

The second worst damaging mode was mode 3, during which tests 5 cracks were sustained; mode 1 was the least damaging, with only 2 cracks occurring.

Considering the clearance mode 2 case, the summary conclusion can only be made that the axial impact threshold level is above 350 m/s^2 (35.7 g) peak. The corresponding secondary impact was 1110 m/s^2 (113.2 g) peak. The calculated impact velocity in this case was about 18 km/h (11 mph) which has previously been identified as the absolute maximum railcar impact velocity that will occur. Limitations of the test rig did not permit testing at higher impact velocities.

Bundles G53601W and G53668W, which had previously undergone fatigue tests at the highest level for 24 h were also tested in the mode 2 configuration. Neither bundle failed after exposure to the combined fatigue and axial impact maximum levels.

(c) Lateral Impact Tests: These tests simulate the vertical component of the shock load which occurs during railcar coupling. The test fixture used in the fatigue and axial impact tests was modified by the addition of a vee-shaped penetrator along its length to produce the appropriate rise time characteristic when dropped into a bed of sand. This apparatus was virtually identical to that used in the hot cell tests/5/, except that an axial clearance of 33 mm was provided. One pair of bundles was tested up to the maximum drop height available. Difficulties were experienced with the fixture guideways which could not be resolved. These limited the maximum velocity at impact and thus the maximum deceleration.

The test results are listed in Table IV. Neither test bundle failed up to a peak deceleration of 390 m/s^2 (39.8 g), although two additional endplate cracks occurred in bundle G53601W. The bundle remained integral. No other defects were found. The time history for the maximum impact is shown in Figure 11. The rise time was 13.0 ms.

Bundle 53601W, it should be noted, was exposed to the maximum levels of all three types of tests - fatigue, axial impact, and lateral impact. This bundle sustained a total of 6 cracks, all in the endplates, as follows: (i) fatigue tests (24 h at 7.35 m/s^2): 2 cracks in the unmonogrammed end (inboard), (ii) axial impact tests (350 m/s^2 peak deceleration): 1 crack in the unmonogrammed end, 1 crack in the monogrammed end, (iii) lateral impact tests (390 m/s^2 peak deceleration): 1 crack in the unmonogrammed end, 1 crack in the monogrammed end. The bundle remained integral even after physical handling on the bench, when it was rolled back and forth several times, then placed vertically on end and rocked laterally ("parallelogrammed").

3.3 Conclusions and Recommendations

3.3.1 Low Temperature Brittle Fracture of Fuel (Task 4)

(a) The fracture toughness of fuel bundle material at 30°C may be lower than at the temperature at which the bundles were tested. However, this effect is judged to be small.

Metallurgical tests may be deferred pending the results of field tests on a prototype transportation system, when actual strength margins can be established.

(b) The change in mechanical strength of bundle elements due to pellet-to-sheath interference over the temperature range of -30 to $+95^\circ\text{C}$ is negligible.

3.3.2 Bundle Impact and Fatigue Tests (Task 8)

(a) The threshold fatigue level for Bruce fuel bundles with an axial clearance of 33 mm is at least 5.20 m/s^2 (0.53 g) rms.

(b) The axial impact threshold level is at least 350 m/s^2 (35.7 g) peak for the worst clearance mode case. The corresponding secondary impact level was 1110 m/s^2 (113.2 g) peak. The secondary impacts which occurred as a result of the axial clearance were always substantially greater by a factor of 3 to 4 than the primary impacts.

(c) Of the three clearance modes tested, the most damaging was mode 2 (all the clearance at the front of the bundle pair).

(d) The lateral impact threshold level with 33 mm axial clearance is at least 390 m/s^2 (39.8 g) peak.

(e) Two bundles were exposed to a combination of the maximum fatigue and axial impact levels. In addition, one of these was also exposed to the maximum lateral impact levels. The fact that neither bundle failed as a result of the combined loading indicates that fuel bundles are in fact very robust when contained in the shipping module geometry.

4.0 MODULE TEST PROGRAM

The module test program, as far as shock and vibration are concerned, was divided into two areas: static loading and dynamic testing (Task 12).

4.1 Module Static Loading (extra task)

The test program was established to determine the effect of frictional forces on the module due to sliding of the fuel bundles within the extent of the axial clearance (33 mm). The details of the work are reported in Reference/9/, and are summarized here. The tests first determined the force exerted on a tube by a fuel bundle as the module is decelerated to rest under both wet and dry conditions. Then this load is applied to the tubes in order to determine the deflection of the module frame and the stress distribution in the tubesheet under this loading. In addition, an assessment was made of the module's ability to resist stresses in a single tube due to overloading, such as may occur if a bundle failed during automated loading.

Under wet conditions, the average force for one bundle at which sliding just occurred was 98 N (24.0 lb), and under dry conditions it was 83 N (18.6 lb). The corresponding coefficients of friction were 0.41 (wet) and 0.34 (dry). For the load deflection test, weights of 22.7 kg (50 lb) per tube (chosen for convenience) were loaded on each row of four vertically oriented module tubes. The strains and deflections were recorded. This was repeated for each of the twelve rows of tubes. By applying the principle of superposition, the individual deflections were summed to arrive at the total deflection assuming a uniform force of 222 N (22.7 kg) per tube. A maximum deflection of 0.38 mm (0.015 in) was calculated on this basis. The maximum recorded stress was under 7.5 MPa (1100 psi).

During the friction tests, it was apparent that the bundles could snag on ridges left inside the tubes due to burn-through of the tube-to-tubesheet welds. This could result in high axial loads on the tubes (and the bundles) if a loading machine were attempting to force a bundle past these obstructions.

In the single tube overload test, a series of weights exerting up to 890 N (200 lb) were applied to one tube. The maximum deflection was only 0.09 mm (0.0035 in) and the maximum stress was 11.7 MPa (1700 psi).

4.2 Module Dynamic Testing (part of Task 12)

This work is reported as part of Reference/10/ and is summarized below.

The structural dynamics of two storage and transportation modules stacked one on top of the other was investigated experimentally in order to define the shock and vibration transmission through the modules. Three-dimensional modal testing of the two module stack identified ten modes of vibration in the frequency range 0 to 1000 Hz which are used to define a model data base for the module stack.

The impact technique was used for the transfer function measurements. The exciter was an instrumented hammer of 5.4 kg mass. The hammer was equipped with a suitable tip to excite the vibration modes in the frequency range of interest (0 to 1000 Hz). The response was measured using sensitive accelerometers. Three responses were measured simultaneously, together with the force signal and processed, on line, for the modal analysis. The data was processed and analyzed using a GenRad 2510 Micro Modal Analyzer.

Figure 12 shows the two storage modules filled with unirradiated CANDU fuel bundles and stacked one on top of the other. Responses were measured in three perpendicular directions (x, y, and z) giving a total of 102 response measurements. The model used in the analysis and the coordinate system are defined in Figure 13. The measurement points are numbered 1 to 34. The excitation F was applied at point 1, in the direction Y as shown in Figure 13.

The transfer functions were computed as a complex function of frequency from averaged Fourier transforms of output accelerations and input force. In order to improve the signal-to-noise ratio each transfer function represents the average of eight measurements. The coherence function was also computed for each measurement point and was used to gauge data quality throughout the measurement program.

Typical results are shown in Figures 14, 15 and 16 for the measurements taken at point 3 in the x, y and z directions respectively. Each transfer function is expressed in terms of its magnitude and phase angle for the frequency range 5 to 1000 Hz.

From examining the transfer functions obtained through the measurement program, the following ten resonances were identified as corresponding to the main modes of vibration:

- (a) 196.791 Hz
- (b) 253.803 Hz
- (c) 315.429 Hz
- (d) 393.350 Hz
- (e) 486.923 Hz
- (f) 518.046 Hz
- (g) 646.335 Hz

- (h) 742.175 Hz
- (i) 859.272 Hz
- (j) 946.819 Hz

The modal analysis was performed using the SDRC MODAL-PLUS-V7.0 software package. To obtain the mode shapes of the module stack, the following technique was used: For each resonance of interest, a measurement (transfer function) where that resonance is clearly visible is selected. A mathematical expression is curve fitted on that measurement for the resonance and the nearest frequency region. The fit gives frequency, damping, amplitude and phase for the resonance. These parameters are then used to give an estimate of amplitude and phase for every measurement. Figure 17 shows the module frame mode shape for mode 1.

This procedure gives for each resonance the amplitude and phase at each measurement. When the geometry of the module stack is known, mode shapes may be generated and displayed so that each point of the module stack moves with the computed amplitude and phase in the given directions. Figure 17 shows the mode shape at 196.8 Hz. The main response occurs due to bending of the top frame members.

4.3 Conclusions and Recommendations

4.3.1 Module Static Loading

(a) The maximum coefficient of static friction occurs under wet conditions and is approximately 0.41. Under dry conditions it is 0.34.

(b) Under a uniform load of 222 N per tube, the displacement of the module at the centre was approximately 0.38 mm (0.015"), and the measured tubesheet stress was less than 7.5 MPa (1100 PSI, 4.4% of yield). According to the ASME Boiler and Pressure Vessel Code, Figure I-9.2, at a stress level such as this, the fatigue life is expected to be in excess of 1 million cycles. Since these 222 N loads are expected to occur only during shunting operations which would not amount to many cycles during the service life of the module, it can be concluded that the fatigue life of the module will exceed the service life. These findings indicate that axial fuel bundle restraint is not necessary to ensure module integrity.

(c) Under a single tube overload condition of up to 890 N (200 lb), the measured tubesheet stress was below 11.7 MPa (1700 psi, 6.8% of yield). Figure I-9.2 of the ASME Boiler and Pressure Vessel Code again indicates a fatigue life in excess of 1 millioncycles. Therefore, single tube overloads up to and in excess of 890 N can be easily accommodated by the module.

(d) A stricter quality control program should be instituted to ensure that the insides of the tubes are free from any obstructions resulting from burn through of the tube-to-tubesheet welds.

4.3.2 Module Dynamic Tests

- (a) The transfer function characteristics of the two module stack has been determined using the modal testing technique.
- (b) The transfer function characteristics can be used to determine the shock and vibration transmission through the modules during road and rail transportation.

5.0 SHOCK AND VIBRATION ENVIRONMENT DURING OFF-SITE TRANSPORTATION (TASKS 11 AND 12)

In order to determine the shock and vibration to which the fuel bundles will be exposed, both the input from the vehicle/tiedown system, as well as the transmission through the module structure are required. The following sections summarize these requirements. Full details are available in Reference/2/.

5.1 Shock and Vibration Input from Vehicle/Tiedown System (Task 11)

Two cases have been examined in detail in reference/2/: (a) vibration response of a tractor-semitrailer/tiedown system and (b) end impact response of a railcar/tiedown system.

5.1.1 Vibration Response of a Tractor-Semitrailer System

A mathematical model has been developed to predict the random response of a tractor - semitrailer system carrying a nuclear fuel shipping cask to road undulations. The model allows the flexibilities associated with the cask, tie-down, and transport frame to be simulated for various design configurations.

The objectives are:

- a) To determine the response of the trailer-cask/tie-down system to the random input of the road profile when the tractor-trailer system is travelling at constant speed.
- b) To determine the levels of cask accelerations for various tractor speeds and road roughnesses.

The tractor-semitrailer model used in this study is shown in Figure 18. It is seen to be two dimensional in the longitudinal and vertical plane. The cab, engine and tractor chassis are considered as one rigid body as is the semitrailer. Both units are allowed to translate in the forward and vertical directions and to pitch, except as constrained by the fifth wheel. The unsprung mass of each wheel and axle assembly is represented by a mass having vertical translational freedom only. The tractor-semitrailer suspension is represented by linear springs and linear viscous dampers. Each of the tires on the tractor and semitrailer

is assumed to be a linear spring and damper having point contact with the road surface.

By considering the constraints imposed by the fifth wheel on the motion of the tractor and semitrailer, the model includes six degrees of freedom: pitching and bouncing motions of the tractor centre of gravity, pitching motion of the semitrailer centre of gravity, and vertical motion and each wheel and axle assembly.

The cask/tiedown/transport frame model is shown in Figure 19. It is seen to be a three-dimensional rectangular plate finite element model, allowing both in-plane and out-of-plane (bending) forces. Elements 1 to 8 represents the transport frame and the bottom plate of the cask, elements 9, 10, 15 and 16 the side walls of the cask, elements 11 to 14 the cask lid, while elements 17 to 20 represent the front and back walls of the cask as well as the tie-down system.

The rectangular plate element allowing in-plane forces is shown in Figure 20. Each node has two degrees of freedom and the forces acting on the nodes lie in the local x-y plane. Figure 21 shows the rectangular plate element allowing out-of-plane (bending) forces.

The modules stack is next considered as a rigid mass having three degrees of freedom x, y, and (nodal point 24) connected through two beams, to nodal points 20 and 8, representing the connection between the modules and the cask, as seen in Figure 22.

Figure 23 shows the complete system model. As seen, the cask/tie-down/transport frame/module system is coupled with the tractor-semitrailer model through two springs of stiffness K each, representing in reality four bolts used to fasten the transport frame to the trailer bed.

It has been recognized from a number of measurements on roadways that their roughness is normally distributed and can be accurately represented as a stationary random process. Spectral densities that closely approximate available experimental data can be described by:

$$\begin{aligned} s(n) &= s(n_0) \left(\frac{n}{n_0}\right)^{-r_1}, n \leq n_0 \\ &= s(n_0) \left(\frac{n}{n_0}\right)^{-r_2}, n \geq n_0 \end{aligned}$$

where $s(n)$ is the spatial density of the road roughness,
[m²/1/m]

$s(n_0)$ is the roughness coefficient (the value of the spectral density at the discontinuity frequency n)

n is the spatial frequency of road roughness

n_0 is $1/2\pi$, [c/m]

r_1, r_2 are waviness

The proposed classification of the properties of road surfaces based on these equations is given in Table 5 and Figure 24.

Figure 25 describes the analysis of the vehicle system excited by road irregularities. The spatial spectrum of the road roughness, $S(n)$, is converted into the temporal spectrum, $S(f)$. Taking into account the time delay caused by the wheelbase, the excitation displacement matrix of all wheelsets, $S(f)$, is obtained. This excitation displacement matrix is calculated in terms of the exciting force matrix, $S(f)$, using the forced stiffness and damping matrices.

The power spectra and the rms values of the vehicle response were calculated and plotted as was the effect of the speed of the vehicle on the root mean square acceleration of the system response. The results are computed for the system parameters given in Table VI. The gross weight of the cask/tiedown/module stack is taken to be 34 545 kg (38 tonnes). Figures 26 and 27 show the configuration used in the analysis for the cask and the support frame respectively. Four road surface qualities as given in Table V are used. For each road condition, and assuming a vehicle forward speed of 80 km/h, the following results are obtained:

- (a) Power spectral density of the longitudinal acceleration at the tiedown connection with the trailer.
- (b) Power spectral density of the vertical acceleration at the tiedown connection with the trailer.
- (c) Power spectral density of the longitudinal acceleration at the tiedown connection with the cask.
- (d) Power spectral density of the vertical acceleration at the tiedown connection with the cask.
- (e) Power spectral density of the longitudinal acceleration of the module stack centre of gravity.
- (f) Power spectral density of the vertical acceleration of the module stack centre of gravity.

In each case the power spectral density is calculated and plotted against road frequency in Hz. Sample results are shown in Figure 28 for road Class D as defined in Table V.

In addition to the above results which were obtained for a vehicle forward speed of 80 km/h, the effect of varying the vehicle speed

on the vertical acceleration of the module stack C.G. and at the tiedown connection with the trailer was evaluated. The results are presented in Figures 29 and 30 for vehicle speeds up to 100 km/h. In general, the acceleration levels increase with increase in speed. Referring to Figure 29, and for a vehicle speed of 100 km/h, the rms value of the vertical acceleration of the module C.G. for road class D is about one order of magnitude higher than the one corresponding to road class A.

The model described includes the flexibilities of the cargo (cask/tiedown/transport frame). This is the main difference from the work described in Reference/10/ and accordingly the acceleration levels are expected to be higher. In fact, they are about 60% higher in average. The levels reported in this study compare, however, more favourably with the results of previous field tests/11/ where an acceleration level of 0.32 g rms was measured in the speed range 80 to 90 km/h.

5.1.2 End Impact Response of a Railcar/Tiedown System

The objective of this study is to determine the shock level to which a railcar-cask-tiedown system is subjected to during end collisions at a speed of 16 km/h (10 mph). The road cask, transportation frame and tiedown configuration as defined in Figures 26 and 27 are again used in this analysis.

Figure 31 shows the analytical model used for a standard 100 ton capacity railcar and coupler, as well as the cask with its tiedown configuration. The model is based on the finite element technique with two dimensional beam elements used to model the vehicle/cask system. Each nodal point possess longitudinal "u", vertical "v", and rotational " θ " degrees of freedom. The moving car is represented by ten beam elements. The tiedown system is represented by element 15. The shipping cask is represented by elements 13, 14, 16, 17, 18 and 19. Elements 11 and 21 represent the connection between the cask and the railcar. The coupler consists of one spring $K_{11,20}$ and damper $C_{11,20}$ as shown in the figure. A hydraulic draft gear with a travel of 254 mm was assumed in the analysis. Mass 20 represents the initially stationary train that the railcar strikes.

A computer program was developed to compute the impact response at selected points within the system. The transient solution was obtained using the Hamming's modified predictor-corrector method which is a modified fourth order Runge-Kutta method. It is important to note that for the transient motion to include the contribution of all the mode shapes present in the mathematical model, the time interval in the integration scheme should be less than the lowest period of free oscillations.

The results of the transient vibration analysis are computed for the set of data given by Table VII. The analysis for end impact was performed for two different combinations of draft gears; hydraulic-hydraulic and friction-hydraulic. The simulation of the two different combinations of draft gears was obtained as follows:

- (a) For the hydraulic-type draft gear, the damping force results from the viscous damping in the coupler; $C_{11,20} \neq 0$ while R is neglected ($R_F = 0$).
- (b) For the case of simulating the friction-hydraulic combination of draft gears, both types of damping forces were considered in the analysis; ie, R_F and $C_{11,20} \neq 0$.

The acceleration time profile calculated at the module C.G. for a railcar carrying a 38-tonne cargo are shown in Figures 32 and 33 for the vertical and horizontal (longitudinal) directions respectively. The corresponding shock spectra are shown in Figures 34 and 35.

As seen in Figure 32, the peak level calculated in the vertical direction is about -3.5 g. Figure 33 shows a peak level of -10 g calculated in the horizontal direction. These results were obtained for the friction-hydraulic combination of draft gears.

The shock spectra defined in Figure 34 for the module C.G. vertical acceleration shows that the majority of the response is concentrated in the 0 to 200 Hz range with the acceleration level reaching 980 m/s (100 g) peak at 170 Hz. The shock excitation in the horizontal direction is also concentrated in the 0 to 200 Hz range (Figure 35).

5.2 Shock and Vibration Transmission through the Module Structure (Task 12)

This section presents the predictions of the actual response of the module tubes to simulated operating inputs. The modal data base (mode shapes and modal parameters) defined in Section 4.2 is used along with the force inputs defined in Section 5.1.1 and 5.1.2.

The principle of computation can be described by looking at the frequency response relationship

$$\frac{X_i}{F_j}(\omega) = H_{ij}(\omega)$$

where i and j are reference and response points respectively, and H is the transfer function. If the frequency response H_{ij} has been measured and the force specified as a function of frequency, the response can be calculated from:

$$X_i = H_{ij}(F_j)$$

For a complete structure this relation can be written in matrix form:

$$\{X\} = \{H\} \{F\}$$

5.2.1 Vibration Response of Module Tubes - Road Transportation Model:

The various components of the module analysis (road transportation model) are given in Figure 36. The power spectral density of the vertical acceleration of the module stack centre of gravity obtained for road class D and a vehicle forward speed of 80 km/h (defined in Figure 28) was used as an input to the module stack modal data base. This input was applied in the vertical direction simultaneously at points 19, 20, 23 and 24 as defined in Figure 13.

Figures 37 and 38 show the response spectra calculated at points 25 and 31 respectively. These responses are obtained in the vertical direction. The corresponding rms values which are the root mean square summation of all the spectral points are 0.73 g and 0.31 g respectively.

The results of computer simulation of the module tube vibration response to road undulations showed a vertical acceleration level of $7.2 \text{ m/s}^2 \text{ rms}$, (0.73 g rms) predicted for the top row and $3.0 \text{ m/s}^2 \text{ rms}$ (0.31 g rms) for the bottom row of the top module. These levels are based on a semitrailer travelling at a speed of 80 km/h (50 mph) on a "class D" or very poor road. At the other road conditions, the vertical responses of the top tubes of the upper module are as follows:

Road Class	A	B	C
Acceleration $\text{m/s}^2 \text{ rms}$	0.59	1.37	3.53
g rms	0.06	0.14	0.36

In contrast, the effect on the acceleration level of vehicle speed on the same type of road is much less pronounced. For example, the reduction in acceleration level on a class D road from 100 km/h to 30 km/h is only 40%. The vibration response is concentrated in the 15 to 450 Hz range (see Figures 37 and 38).

5.2.2 Shock Response of Module Tubes - Railcar End Impact Model:

The various components of the module analysis (railcar end impact model) are given in Figure 39. The acceleration frequency profiles obtained for a railcar carrying a 38-tonne cask and striking a 226.8-tonne stationary train at 16 km/h, as defined in Figures 34 and 35 for the vertical and longitudinal (horizontal) directions were used as excitation inputs in this case.

For the vertical direction, the excitation defined in Figure 34 was applied simultaneously at points 19, 20, 23 and 24 (see Figure 13) and the response calculated in the vertical direction at points 25 and 31. The results are shown in Figure 40 and 41.

Figures 42 and 43 show the results obtained due to the excitation defined in Figure 35. In this case the response is examined in the horizontal direction at points 25 and 31 respectively.

The acceleration frequency profiles presented in this section were calculated for the 0 to 1000 Hz frequency range. The displacement associated with frequencies higher than 1000 Hz was assumed to be minimal.

The railcar coupling analysis shown in Figures 40 to 43 indicates a shock response reaching a maximum of 686 m/s^2 peak (70 g peak) at 850 Hz for the top row and approximately half this magnitude for the bottom row of the top module. These shock spectra are in reasonable agreement with data published elsewhere in the literature.

5.3 Conclusions

- 5.3.1 The random response of a tractor-semitrailer system carrying an irradiated nuclear fuel shipping cask has been analytically determined. The results are in reasonable agreement with published experimental data.
- 5.3.2 An increase in road vehicle speed produces higher acceleration levels. Over a speed range of 20 to 100 km/h (12.5 to 62.4 mph) the maximum increase for any road is about 40%. Travel on very poor roads (class D) results in an increase of rms vertical acceleration of the cask (and internals) of about one order of magnitude in comparison with smooth roads (class A).
- 5.3.3 The vibration transmission through the two module stack structure during road transportation has been determined using the modal testing technique.
- 5.3.4 The simulation of the vibration environment during road transportation indicates that vertical preload of the module stack is not necessary.
- 5.3.5 The shock response to a coupling impact of a railcar carrying a road cask has been determined by analytical modelling.
- 5.3.6 For the railcar coupling situation, the predicted deceleration levels indicate that relative motion between the module stack and the cask will occur. Since the clearances between the module frame and the vertical guides and the lid are reasonably small (about 6 mm per side horizontally and 12 mm vertically) large excursions and hence loads should not occur. The present analytical model cannot predict the response due to this motion as it is linear. An appropriate non-linear model would be extremely complex and financially prohibitive. A practical approach to the problem is to measure this behaviour and its effects on fuel bundles during prototype transportation system trials.

5.4 Recommendations

5.4.1 The design of the cask need not provide for module stack preload for road transport.

5.4.2 The results presented in this section should be taken in the context of the limitations imposed by mathematical modelling. Actual field measurements on the prototype transportation system are recommended to verify the analytical modelling and to determine those effects which cannot be practically modelled.

6.0 SUMMATION OF RESULTS

The results of each of the component parts of the Phase II Shock and Vibration Program have been discussed in the appropriate sections of this report. The sections are listed below for convenience.

Task:	4	8	11	12	extra task
Section:	3.3.1	3.3.2	5.1	5.2	4.3.1

In order to assess the overall performance of the transportation system, these tasks will now be combined to determine:

- (a) Fuel bundle survivability during transportation (Task 14), and
- (b) Shipping module integrity during transportation.

6.1 Fuel Bundle Survivability

The results of Task 4 suggest that the fracture toughness of fuel bundle material (and hence bundle impact and fatigue limits) may be reduced when the bundle temperature is -30°C . Even though there is some evidence and considerable opinion that this reduction may only be slight/8/, it would be of academic interest to verify this by actual experiment. The effect of temperature on bundle element strength due to pellet-to-sheath clearance over the range of -30 to $+95^{\circ}\text{C}$ has been shown to be negligible.

Given the known data, an assessment of fuel bundle survivability during transportation can be made. Assessments for vibration and shock will be made separately.

6.1.1 Fuel Bundle Vibration

The highest module tube vibration levels during road transport have been shown to occur at the top row of the upper module. The magnitude varies with the vehicle speed and road roughness. At a typical speed of 80 km/h (50 mph) on a Class D road (very poor), the vibration input to the bundles in the top row is 7.2 m/s^2 (0.73 g) rms. This is clearly above the bundle fatigue threshold

level of 5.20 m/s^2 (0.53 g) rms, and thus continuous operation under these conditions could not be permitted. However, operation on Class C (poor) roads or better, which most Provincial highways are expected to be, would not present any hazard to fuel bundle integrity.

The vibration levels occurring during rail transport were not addressed in the Phase II program because of budgetary constraints. However, the results of the Phase I analysis can be used for comparative purposes. Here the maximum calculated vibration level was 3.12 m/s^2 (0.32 g) rms in the critical vehicle speed range (29-35 km/h), well under the 5.20 m/s^2 (0.53 g) rms threshold. The improved modelling in the case of the tractor-semitrailer increased the predicted vibration level by about 60%. If the same were to apply in the rail transportation case, the hypothetical level would then be 5.02 m/s^2 (0.51 g) rms, very close to the threshold. In this case, prolonged operation at 29 to 35 km/h (18 to 22 mph) should be avoided.

6.1.2 Fuel Bundle Shock

The maximum shock response to railcar coupling occurs in the top row of tubes of the upper module in both the horizontal-axial and vertical directions. (The modelling does not consider the horizontal-transverse case since the model is two dimensional in the x-y plane as shown in Figure 31. The forces in the horizontal-transverse plane are judged to be significantly lower).

The maximum peak magnitude in both directions is 686 m/s^2 (70 g) peak, in the vertical case at about 900 Hz and in the horizontal case at about 850 Hz. Although the corresponding displacements are low (0.05 mm or 2 mils p-p) the resulting loads are still being generated. However, only in the vertical direction does this transfer to the fuel bundle. In the horizontal-axial direction load is applied only through friction; hence there is relative motion between the bundle and the tube.

(i) For the horizontal case, it is assumed there is no rigid body motion of the module stack with respect to the cask (a requirement of the linear model). Thus, the loads acting on the bundles consist of axial friction forces, initially, as the cask and module decelerate and the bundles slide to the extent of the axial clearance. Then the bundles experience their main axial impact as they contact the cask wall. The cask and module deceleration correspond to the "initial" impact of the axial impact tests, whereas the impact of the bundles on the cask wall corresponds to the "secondary" impact. The analytical model does not consider secondary impact case.

The peak horizontal deceleration of the cask/module CG (Figure 33) is predicted to be 98.1 m/s^2 (10.0 g), well under the maximum of 350 m/s^2 (35.7 g) achieved in the tests. It should be noted that the impact margin is actually greater than that indicated, since the tests were terminated before bundle failure could be achieved.

(ii) For the vertical case, the results of analysis indicate that an acceleration of 686 m/s^2 (70 g) peak is imposed on the fuel bundles by the top row of module tubes. It should be noted that this level applies only to location 25 at a particular instant of time. At the same instant, the acceleration levels at other points along the tube are of different magnitude and phase relative to location 25. In other words, the shock response of the module tube is a vibration transient, as opposed to rigid body acceleration applied simultaneously all along the length of the tube.

In the lateral impact tests, the simulated module tube within the rigid test fixture behaves in a manner similar to that just described, ie as a vibration transient. However, two parameters are not matched in exact detail between the analytical and test cases: a) the input time histories and b) the transfer functions between the point of impact and the bundles. Nevertheless, certain basic similarities exist:

a) input time histories: a comparison of Figures 11 (test) and 32 (analysis) shows that the time from initiation to the time at which the first deceleration peak occurs is the same, approximately 8.0 ms. In both cases, additional peaks occur over a total duration of about 50 ms.

b) transfer functions: An experimental modal analysis was done on the lateral impact test fixture in Reference /5/. The transfer function between the fixture base and the simulated module tube (mid-span location) in the vertical plane is given in Figure 44. When compared with the vertical transfer function for the module stack (Figure 16), it can be seen that the frequencies at which peak responses occur are in good agreement, particularly at frequencies above 300 Hz. These are tabulated below:

Module:	175	235	380	490	550	610	680	900
Fixture:	150	250	365	490	560	605	682	890

Thus, the effect on the fuel bundles of the test impact time history can be reasonably and justifiably compared with the effect produced by the predicted time history given in Figure 32. The predicted peak level of 34.3 m/s (3.5 g) is about one order one order of magnitude lower than the maximum peak produced by the test impacts (390 m/s^2 , 39.8 g), which did not result in bundle failure.

6.2 Shipping Module Integrity During Transportation

6.2.1 Shipping Module Vibration

The ability of the shipping module to survive the vibration (fatigue) effects during transportation was not addressed in Phase II, since it had been adequately covered in Phase I/1/. The single-tube fixture used in the fatigue tests was fabricated using the same weld and material details as the shipping module. No

evidence of fatigue failure occurred during the fatigue testing program which exposed the fixture to 72 h of continuous vibration at varying levels up to 5.2 m/s (0.53 g) rms and 0.5 h at 13.9 m/s (1.4 g) rms. On this basis, the module should be capable of withstanding the fatigue effects of several transportation cycles, if required.

6.2.2 Shipping Module Shock

Tests were carried out to determine the effects of loads on the tubesheets caused by rail car coupling loads that accelerate the bundles along the tube. Maximum loads occur just before the bundle begins to slide.

The static coefficient of friction between the tube and the bundle, under dry and moist conditions was determined to range between 0.3 and 0.4. Module tubes were then loaded, axially, to 222N each, corresponding to 0.4 times the weight of two bundles. The measured strains indicated a maximum stress in the tubesheet of less than 5 percent of yield. Single tube overloads were simulated with loads up to 890 N, resulting in maximum stresses being less than 7 percent of yield. The stresses in the single tube overload were found to be reasonably localized about the loaded tube.

These results can be extrapolated to show that stresses throughout the module are below yield under the inertial effects of the bundles due to rail car coupling impact, where axial decelerations cause the bundles to slide in the tubes, and where the effective weight of the bundles increases due to vertical acceleration.

Bending of the module frame under rail car coupling impact was also checked, assuming the module is fixed and an axial load is applied uniformly. Stress are conservatively estimated and found to be well below yield. Bending of the module frame is not predicted under normal transportation conditions.

7.0 SUMMARY OF CONCLUSIONS AND RECOMMENDATIONS

7.1 There is ample evidence that the fuel bundles, when shipped with an axial clearance up to 33 mm (1.31 in), will survive both the shock and vibration transportation environments. The minimum margins of safety based on each of the predicted environments are given below:

i) road transport, (Class C road at 80 km/h):	1.5
ii) rail transport (Phase I data):	1.7
iii) rail car end impact - vertical:	11.4
- horizontal:	3.6

7.2 For rail car end impacts, the coupler must have a long enough stroke that bottoming-out does not occur (a minimum travel of 255 mm (10 inches) is recommended in reference/2/).

7.3 For rail transport, prolonged operation in the critical speed range of 29 to km/h (18 to 22 mph) should be avoided.

7.4 For road transport, prolonged operation on Class D (very poor) roads should be avoided.

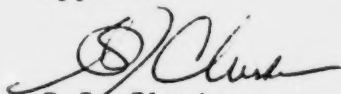
7.5 The shipping and storage module is adequately designed for fuel bundle transport.

7.6 The results of the Shock and Vibration Program should be taken in the context of the limitations imposed by mathematical modelling. Actual field measurements on the prototype transportation system are strongly recommended to verify the analytical modelling and to determine those effects which cannot be practically modelled.

ACKNOWLEDGEMENTS

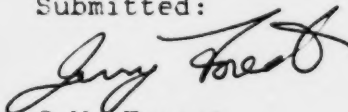
The author acknowledges the contributions by Dr. M.A. Elbestawi and Mr. D.F. Lau for reference/2/, excerpts from which appear in this report and by Mr. G.M. Smrke for reference/9/. Also fruitful discussions with Drs. M.A. Elbestawi, J.D. Tulk and M. Leger of Research Division and Messrs R.A. Eager and B.P. Dalziel of Design & Development Division - Generation are gratefully acknowledged.

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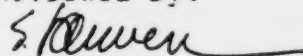
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8. Memorandum, L.A. Simpson to A.D. Lane, "Fracture Behaviour of Zirconium Alloys at Sub-Freezing Temperatures", AECL-WNRE File No MSB-81-025, 1981/02/06.
9. "Irradiated Fuel Shipping Module Transportation Tests", G.M. Smrke, Ontario Hydro Research Division Report 83-321-K, 1983/09/20.
10. "Random Response of a Tractor - Semitrailer System Carrying an Irradiated Nuclear Fuel Shipping Flask", M.A. Elbestawi and M.A. Dokainish, Ontario Hydro Research Division Report 80-209-K, 1980/06/12.
11. "Irradiated Fuel Transportation Shock and Vibration Program Rail and Truck Field Tests", J.W. Forest, Ontario Hydro Research Division Report 79-149-K, 1979/03/29.
12. Memorandum, L.A. Simpson to A.D. Lane, AECL-WNRE File Number MSB-81-025, 1981/02/06.

TABLE I
FUEL BUNDLE ALLOCATIONS

BUNDLE S/N	HYDRIDE PROCEDURE DEVELOPMENT	FATIGUE TESTS	AXIAL IMPACT TESTS	LATERAL IMPACT TESTS	REMARKS:
G53667W	X	-	-	-	
G53685W	-	X (4.90 m/s ²)	-	-	Also Tested at 19.6 m/s ² peak
G53663W	-	X (4.90 m/s ²)	-	-	
G53687W	-	X (6.13 m/s ²)	-	-	
G53600W	-	X (6.14 m/s ²)	-	-	
G53601W	-	X (7.35 m/s ²)	X Mode 2	X	Extra axial test.
G53668W	-	X (7.35 m/s ²)	X Mode 2	-	
G55665W	-	-	-	X	
G53597W	-	-	X Mode 1	-	
G53686W	-	-	X Mode 1	-	
G53670W	-	-	X Mode 2	-	
G53599W	-	-	X Mode 2	-	
G53598W	-	-	X Mode 3	-	
G53664W	-	-	X Mode 3	-	

TABLE II
FATIGUE TESTS

TEST #	BUNDLE FRONT	S/N REAR	VIBRATION LEVEL [m/s ²]	(rms) [g]	TEST DURATION [h]	REMARKS:
1	G53685W	G53663W	3.47	0.35	2.5	no cracks
					6.0	no cracks
					8.0	no cracks
					10.0	no cracks
					12.2	no cracks
					14.0	no cracks
					16.0	no cracks
					18.0	no cracks
					20.0	no cracks
					22.0	no cracks
					24.0	no cracks. Bundles integral
2	G53687W	G53600W	4.33	0.44	2.0	no cracks
					4.0	fretting wear on wear pads rear end of G53600W
					6.0	G53600W rotated 90°
					8.0	wear on both endplates of G53600W
					9.0	wear on rear endplate of G53687W
					10.0	no change
					12.0	no change
					14.0	no change
					15.75	2 cracks on rear endplate of G53687W
					17.0	no change
					18.0	significant wear on rear wear pads of G53600W
					19.0	significant wear on front wear pads of G53687W
					20.0	no change
					21.0	1 crack on front endplate and 3 cracks on rear endplate of G53687W
					22.0	wear on front ends of elements of G53687W
					23.0	
					24.0	6 cracks total on G53687W. No cracks in G53600W. Bundles integral
3	G53601W	G53668W	5.20	0.53	1.0	
					3.2	
					4.0	initial wear on rear endplate of G53668W
					8.0	no cracks
					10.0	no cracks
					12.0	no cracks G53601W rotated to 45° position
					14.0	G53668W rotated to 135°, G53601W to 330° position
					16.0	G53601W rotated to 30° position
					18.0	no cracks
					20.0	G53668W rotated to 15°, G53601W to 315° position
					22.3	no cracks
					24.0	G53668W rotated to 135° position, 2 cracks on rear endplate of G53601W. Bundle integral
4	G53685W	G53663W	13.87	1.41	0.5	no cracks. Bundle integral

TABLE III
AXIAL IMPACT TESTS

TEST #	DROP (cm)	HEIGHT (in)	CALCULATED IMPACT VELOCITY (m/s)	MAXIMUM DECELERATION (m/s ²)	MAXIMUM DECELERATION (peak) (g)	IMPACT DURATION (ms)	CLEARANCE MODE	BUNDLE FRONT	B/W REAR	REMARKS:
1	15.9	6.25	2.15	- 110. + 540.	- 11.2 + 55.1	10.0 34.0	1	G53597W	G53686W	
2	29.2	11.25	2.66	- 80. + 930.	- 8.2 + 94.8	8.0 24.4	1	G53597W	G53686W	
3	41.3	16.25	3.10	- 175. + 930.	- 17.8 + 84.6	12.2 26.7	1	G53597W	G53686W	
4	54.0	21.25	3.48	- 220. + 800.	- 22.4 + 81.6	9.6 23.5	1	G53597W	G53686W	
5	66.7	26.25	3.82	- 240. + 920.	- 24.5 + 93.8	11.7 23.0	1	G53597W	G53686W	2 endplate cracks at front end of G53686W
6	90.8	35.75	4.39	- 270. +1060.	- 27.5 +108.1	12.0 21.8	1	G53597W	G53686W	no further cracks found
7	103.5	40.75	4.67	- 320. +1380.	- 32.6 +140.7	11.5 20.9	1	G53597W	G53686W	no further cracks found
8	125.1	49.25	5.10	- 390. +1285.	- 39.8 +131.0	11.0 20.0	1	G53597W	G53686W	2 cracks total. Bundle integral.
1	16.5	6.5	2.18	- 120. + 510.	- 12.2 + 52.0	11.0 31.8	2	G53670W	G53599W	
2	28.3	11.13	2.65	- 120. + 575.	- 12.2 + 58.6	8.2 28.9	2	G53670W	G53599W	
3	42.5	16.75	3.14	- 175. + 700.	- 17.8 + 71.4	8.9 24.0	2	G53670W	G53599W	
4	52.7	20.75	3.44	- 190. + 715.	- 19.4 + 72.9	10.0 22.7	2	G53670W	G53599W	2 endplate cracks at front end of G53599W
5	66.7	26.25	3.82	- 220. + 770.	- 22.4 + 78.5	8.9 21.4	2	G53670W	G53599W	2 additional cracks on G53599W 1 endplate crack at rear of G53670W
6	92.7	36.5	4.44	- 275. +1010.	- 28.0 +103.0	8.9 19.4	2	G53670W	G53599W	1 additional crack on front of G53599W
7	102.9	40.5	4.66	- 280. + 760.	- 28.6 + 77.5	10.8 19.0	2	G53670W	G53599W	increased friction on swing arm (bolts tightened)
8	117.5	46.25	4.95	- 300. + 880.	- 30.6 + 89.7	8.8 18.3	2	G53670W	G53599W	1 endplate crack at front end of G53670W
9	112.4	44.25	4.85	- 325. + 970.	- 33.1 + 98.9	8.4 17.5	2	G53670W	G53599W	1 endplate crack at rear end of G53670W 8 cracks total. Bundle integral
1	16.5	6.5	2.18	- 85. + 450.	- 8.7 + 45.7	11.7 34.8	3	G53664W	G53598W	
2	28.9	11.38	2.68	- 125. + 615.	- 12.7 + 62.7	9.9 30.5	3	G53664W	G53598W	
3	42.5	16.75	3.14	- 130. + 680.	- 13.3 + 69.3	16.0 30.0	3	G53664W	G53598W	
4	54.0	21.25	3.48	- 185. + 905.	- 18.9 + 92.3	10.4 23.8	3	G53664W	G53598W	
5	69.2	27.25	3.88	- 220. + 840.	- 22.4 + 85.7	9.2 22.0	3	G53664W	G53598W	1 endplate crack at front end of G53598W
6	94.6	37.25	4.48	- 265. + 940.	- 27.0 + 95.9	8.7 20.0	3	G53664W	G53598W	1 endplate crack at rear end of G53664W
7	105.4	41.5	4.71	- 285. +1115.	- 29.1 +113.7	8.9 19.3	3	G53664W	G53598W	
8	116.2	45.75	4.93	- 325. +1115.	- 33.1 +113.7	9.0 18.7	3	G53664W	G53598W	1 endplate crack in rear end of G53664W
9	125.7	49.5	5.11	- 325. +1230.	- 33.1 +125.4	8.9 18.1	3	G53664W	G53598W	2 endplate cracks on front end of G53598W 5 cracks total. Bundle integral
1	54.3	21.38	3.48	- 210. + 460.	- 21.4 + 46.9	9.2 22.5	2	G53668W	G53601W	1 endplate crack on rear end of G53598W
2	96.8	38.13	4.53	- 270. +1070.	- 27.5 +109.1	9.0 19.9	2	G53668W	G53601W	1 endplate crack on front end of G53598W 1 endplate crack on rear end of G53668W
3	120.0	47.25	5.00	- 350. +1110.	- 35.7 +113.2	9.2 19.0	2	G53668W	G53601W	1 endplate crack on front end of G53598W. 2 endplate cracks on rear end of G53668W 6 cracks total. Bundle integral

THIS BUNDLE HAD TWO CRACKS IN OUTBOARD END FROM PREVIOUS FATIGUE TESTS.

TABLE IV

LATERAL IMPACT TESTS

TEST #	DROP [cm]	HEIGHT [in]	MAXIMUM DECELERATION		IMPACT RISE TIME [ms]	BUNDLE S/N		REMARKS:
			[m/s ²]	[g]		FRONT	REAR	
1	41.3	16.25	102.5	10.5	19.0	G53601W*	G53665W	no change
2	71.8	28.25	180.0	18.4	10.2	G53601W	G53665W	no change
3	101.6	40.0	202.5	20.6	10.0	G53601W	G53665W	no change
4	132.1	52.0	229.5	23.4	8.5	G53601W	G53665W	no change
5	162.6	64.0	329.0	33.5	14.0	G53601W	G53665W	1 endplate crack in inboard end of G53601W, 1 endplate crack in outboard end. No cracks in G53665W.
6	187.5	73.8	390.0	39.8	13.0	G53601W	G53665W	no change 2 new cracks, 6 in total. Bundles integral.

* THIS BUNDLE HAD 2 ENDPLATE CRACKS IN OUTBOARD ENDPLATE FROM PREVIOUS FATIGUE TESTS, AND 1 CRACK ON OUTBOARD END AND 1 CRACK ON INBOARD END FROM AXIAL IMPACT TESTS.

TABLE V

ROAD SURFACE CHARACTERISTICS

Road Class	$s(n_o)$	$s(n_o)$	r_1	r_2
	Range	Geometric Mean		
A Good	8 - 32	16	2	1.5
B Average	32 - 128	64	2	1.5
C Poor	128 - 512	256	2	1.5
D Very Poor	512 - 2048	1024	2	1.5
$s(n_o)$ is measured in units of $10^{-6} \text{ m}^2/\text{l/m}$				

TABLE VI

BASE LINE VALUES FOR ARTICULATED VEHICLE MODEL1. TRACTORa. General

Sprung Mass (M_t)	9600 kg
Front unsprung ^t mass (M_1)	360 kg
Rear upsprung mass (M_2)	1450 kg
Pitch moment of inertia (I_t)	3390 kg·m ²

b. Dimensions

b_1 2.25 m	b_5 1.4 m
b_2 2.05 m	a_1 -0.3 m
	a_2

2. SEMITRAILERa. General

Sprung Mass (M_s)	9900 kg
Unsprung Mass ^s (M_3)	1450 kg
Pitch moment of inertia (I_s)	152 000 kg·m ²

b. b_3 3.96 m
b_4 6.00 m
a_2 0.9 m

3. SUSPENSION CHARACTERISTICS

K_1 357 kN/m	C_1 11.5 kN·s/m
K_2 630 kN/m	C_2 29.0 kN·s/m
K_3 630 kN/m	C_3 29.0 kN·s/m

4. TIRE CHARACTERISTICS

K_{t1} 1560 kN/m	C_{t1} 0.7 kN·s/m
K_{t2} 5250 kN/m	C_{t2} 1.2 kN·s/m
K_{t3} 5250 kN/m	C_{t3} 1.2 kN·s/m

5. VEHICLE SPEED

80 km/h

TABLE VII

BASELINE VALUES FOR THE RAILCAR-CASK-TIEDOWN MODEL

Cask/tiedown/transport frame/module stack weight:

38 tonnes (41.9 Tons)

Railcar Weight: 27.2 tonnes (30 Tons)

Stiffness of Draft Gear $K_{11,20} = 16,462 \text{ N/m}$ (94,000 lb/in)

Viscous Damping in the Draft Gear $C_{11,20} = 210 \text{ 150 N/m/s}$
(1200 lb/in/s)

Stationary Train Weight = 226.8 tonnes (500 000 lb)

Friction Force $R_F = (\text{coefficient of friction of } 0.2) =$
 $4.448 \times 10^5 \text{ N}$ (100 000 lb)

Constant of Proportional Damping $\alpha = \beta = 0.02$

$V_a = 16 \text{ km/h}$ (10 mph)

$E = 2.0684 \times 10^{11} \text{ Pa}$ ($30 \times 10^5 \text{ psi}$)

$p = 0.3$

$\Delta t = 10 \text{ } \mu\text{s}$ (time step in the integration scheme)

Length of Railcar = 13.182 m (519 in)

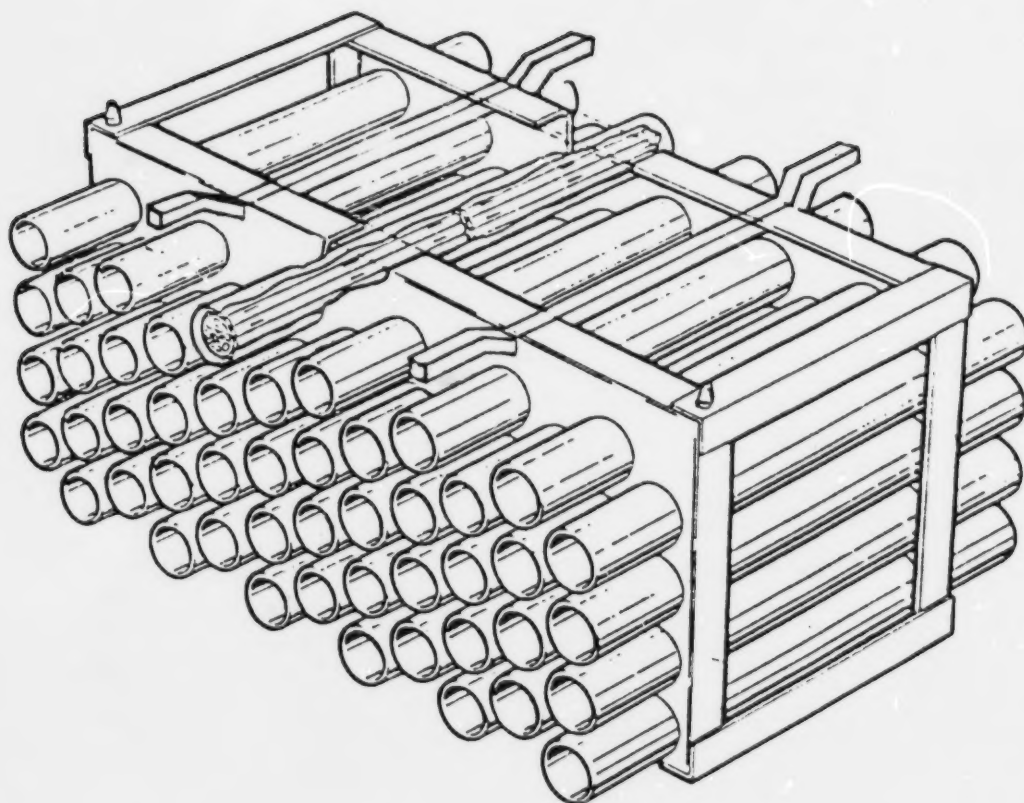


FIGURE 1
IRRADIATED FUEL SHIPPING MODULE

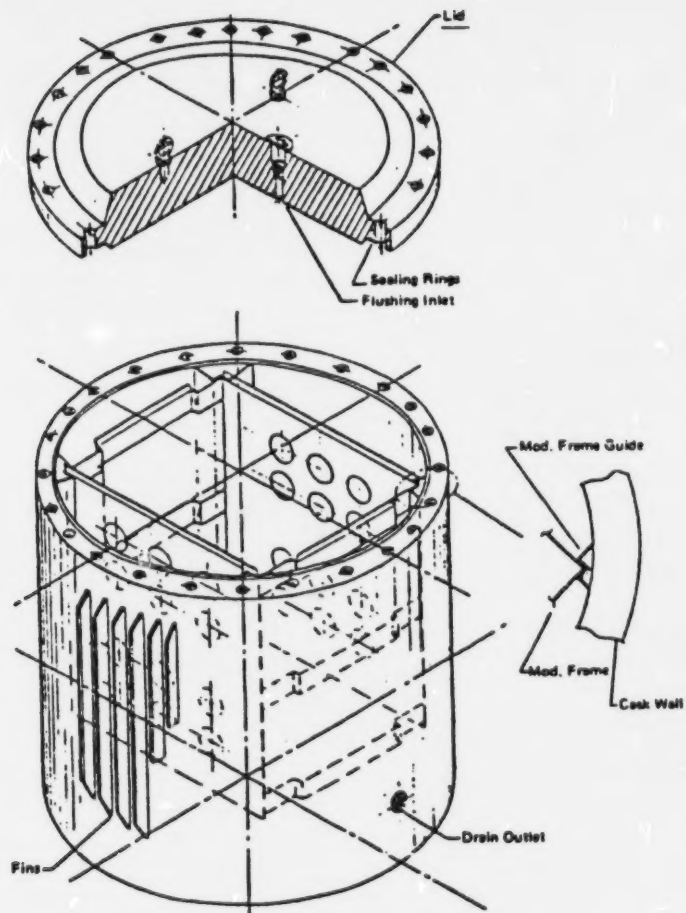


FIGURE 2
CASK CONCEPT - CIRCULAR

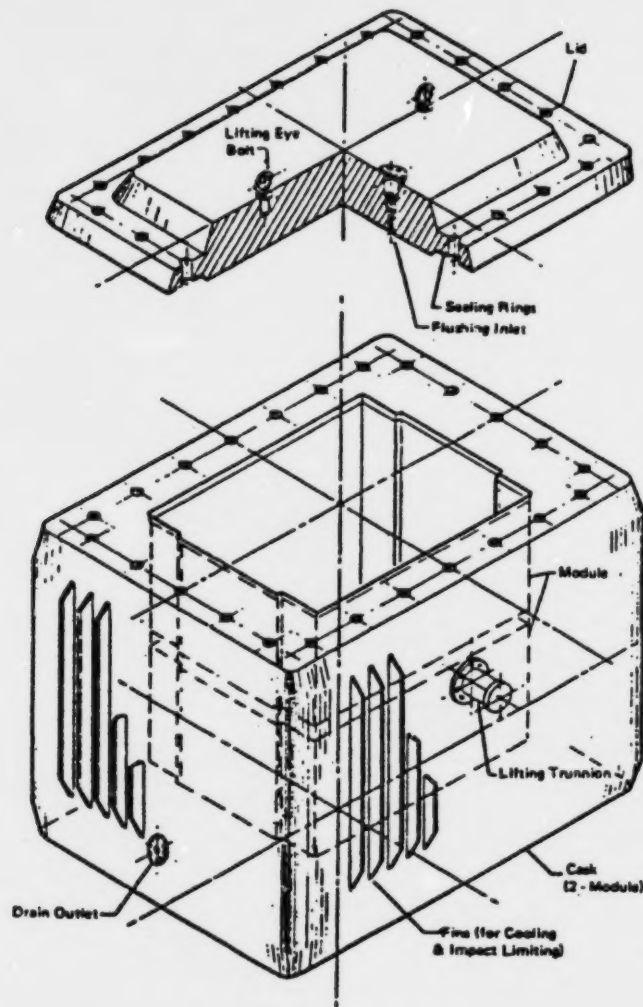


FIGURE 3
CASK CONCEPT - RECTANGULAR

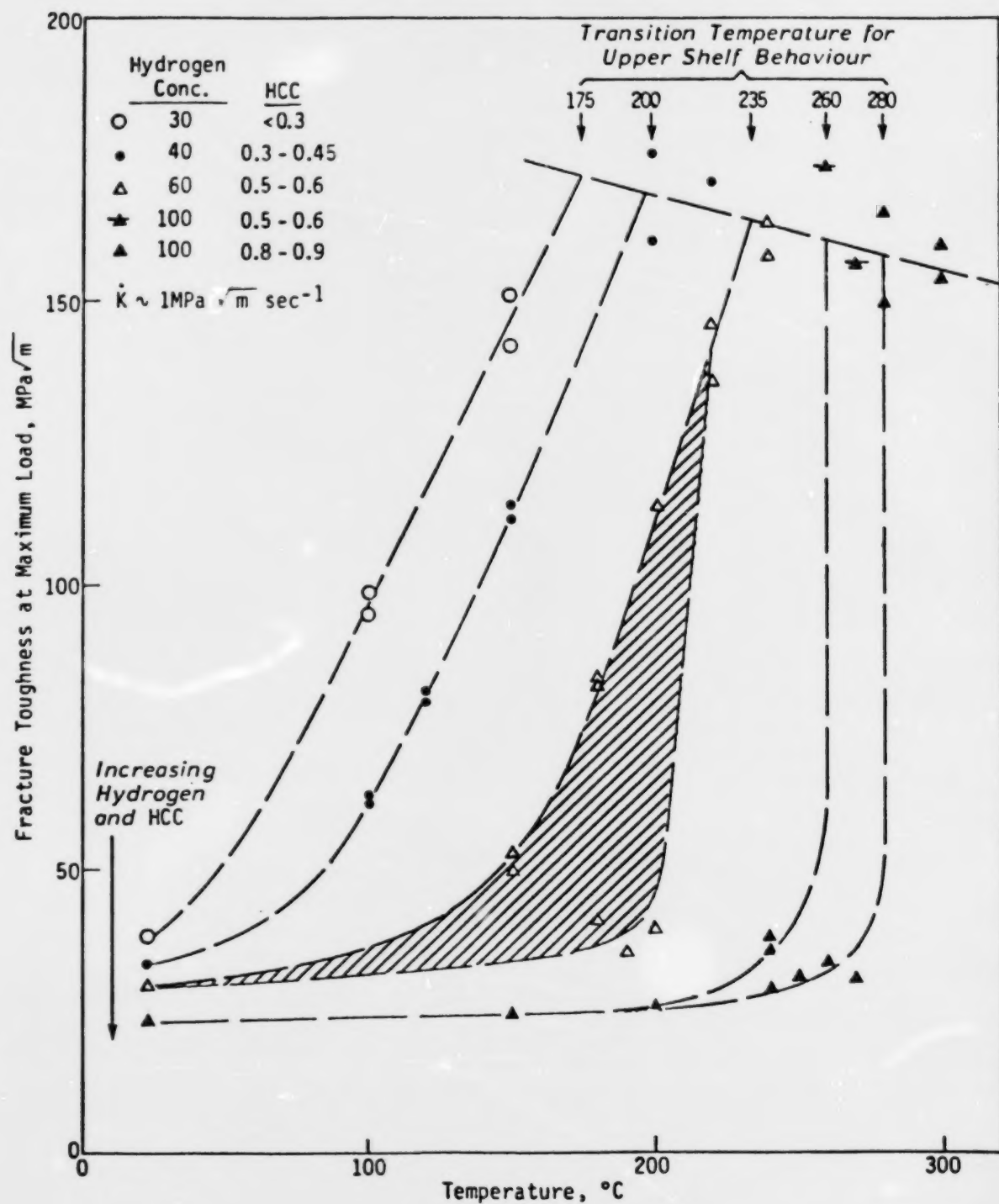


FIGURE 4
FRACTURE TOUGHNESS VERSUS TEMPERATURE FOR
ZIRCALOY-2 COMPACT TENSION SPECIMENS WITH
RADIAL HYDRIDES, 30 PPM TO 100 PPM HYDROGEN

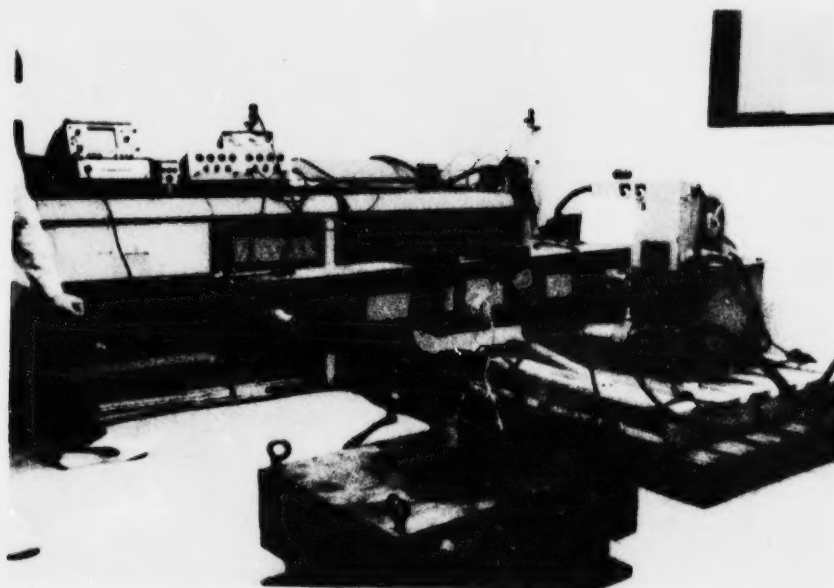
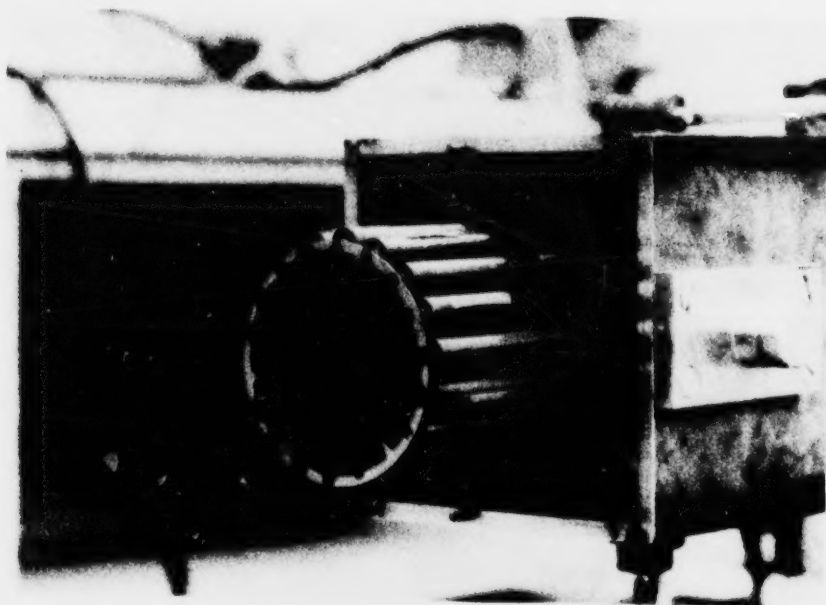


FIGURE 5

FATIGUE TEST CONFIGURATION



BUNDLE G53687W

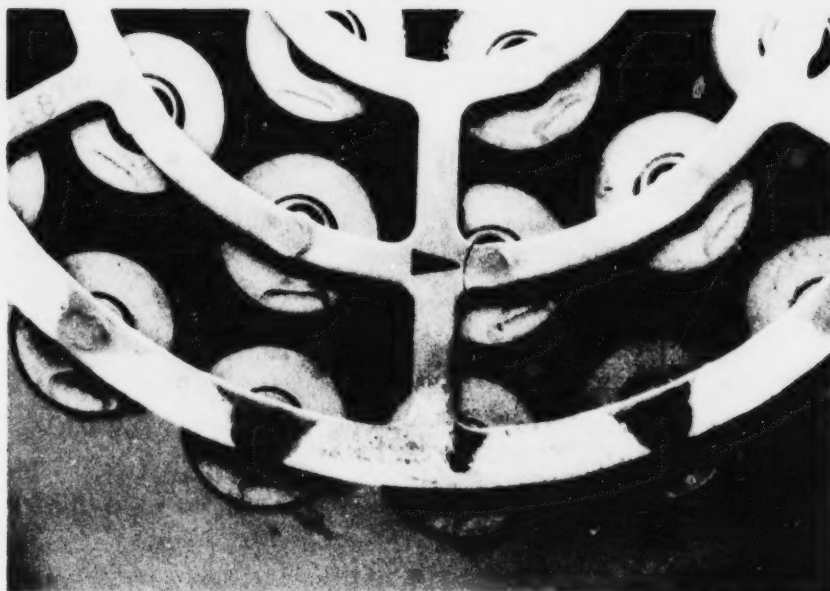
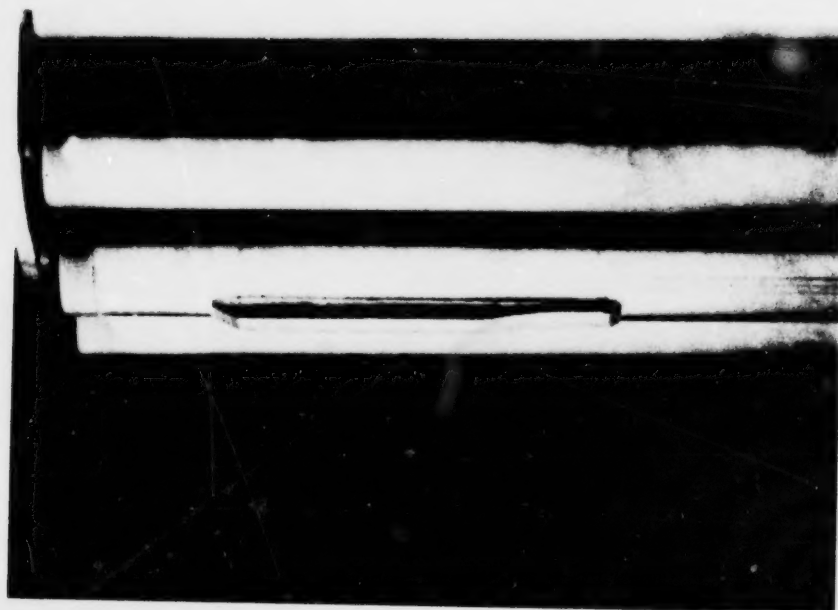
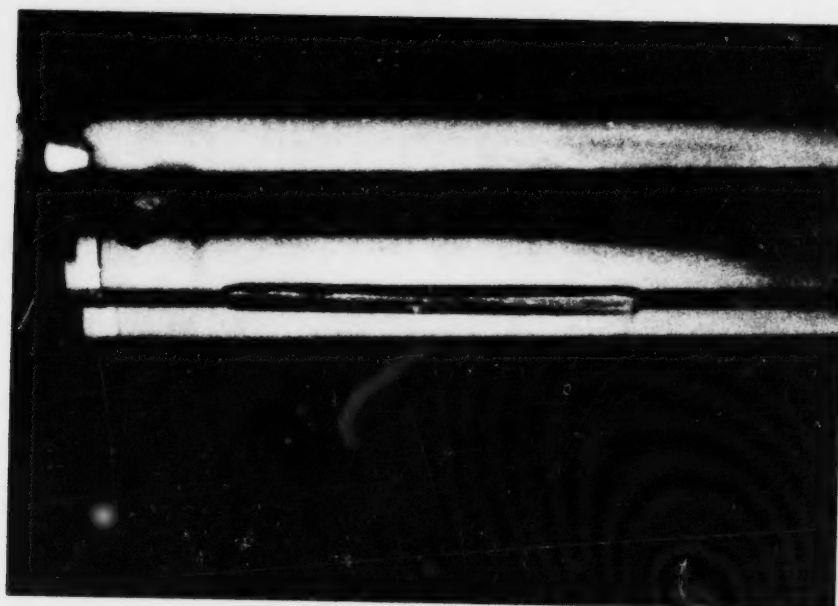


FIGURE 6

TYPICAL FATIGUE CRACKS



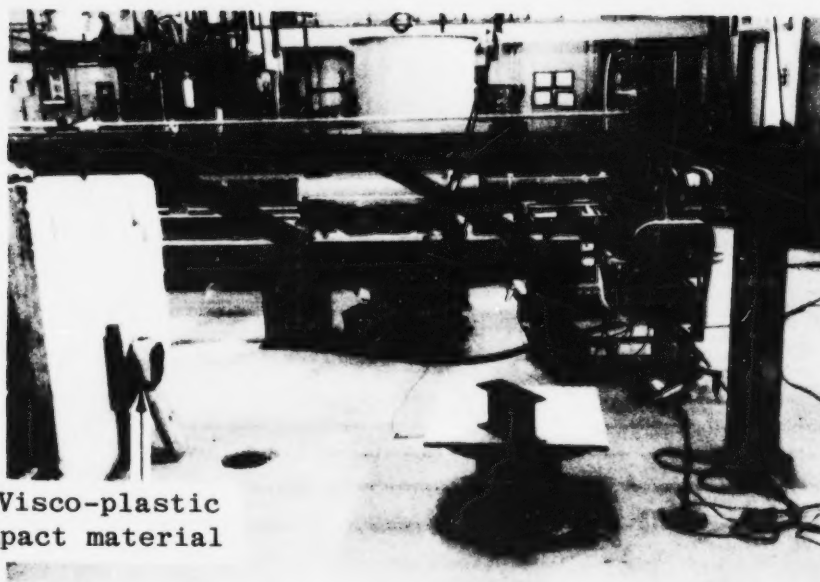
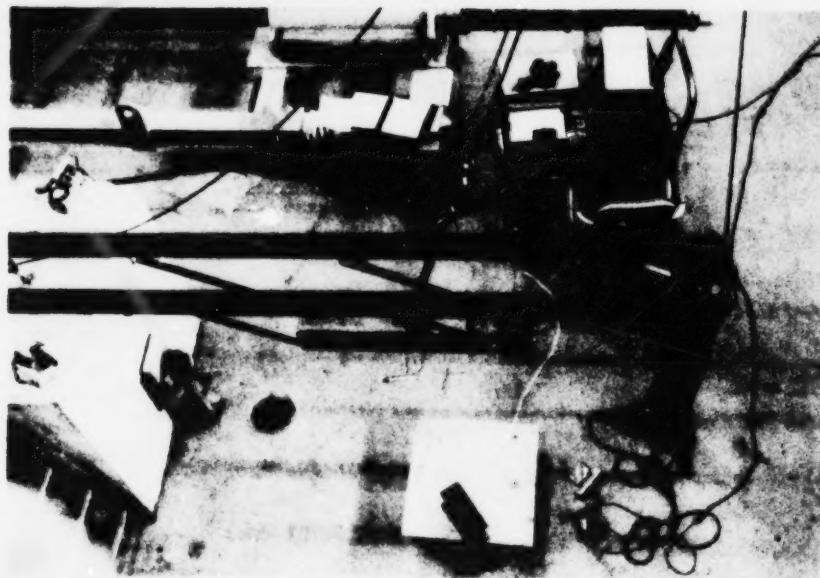
7a



7b

FIGURE 7

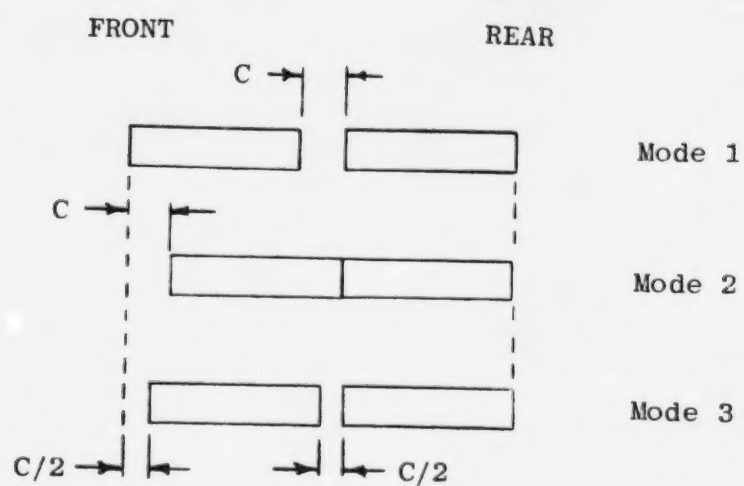
WEAR PATTERNS ON BUNDLE WEAR PADS



Visco-plastic
impact material

FIGURE 8

AXIAL IMPACT TEST APPARATUS



$C = 33\text{mm}$

FIGURE 9
AXIAL TEST CLEARANCE MODES

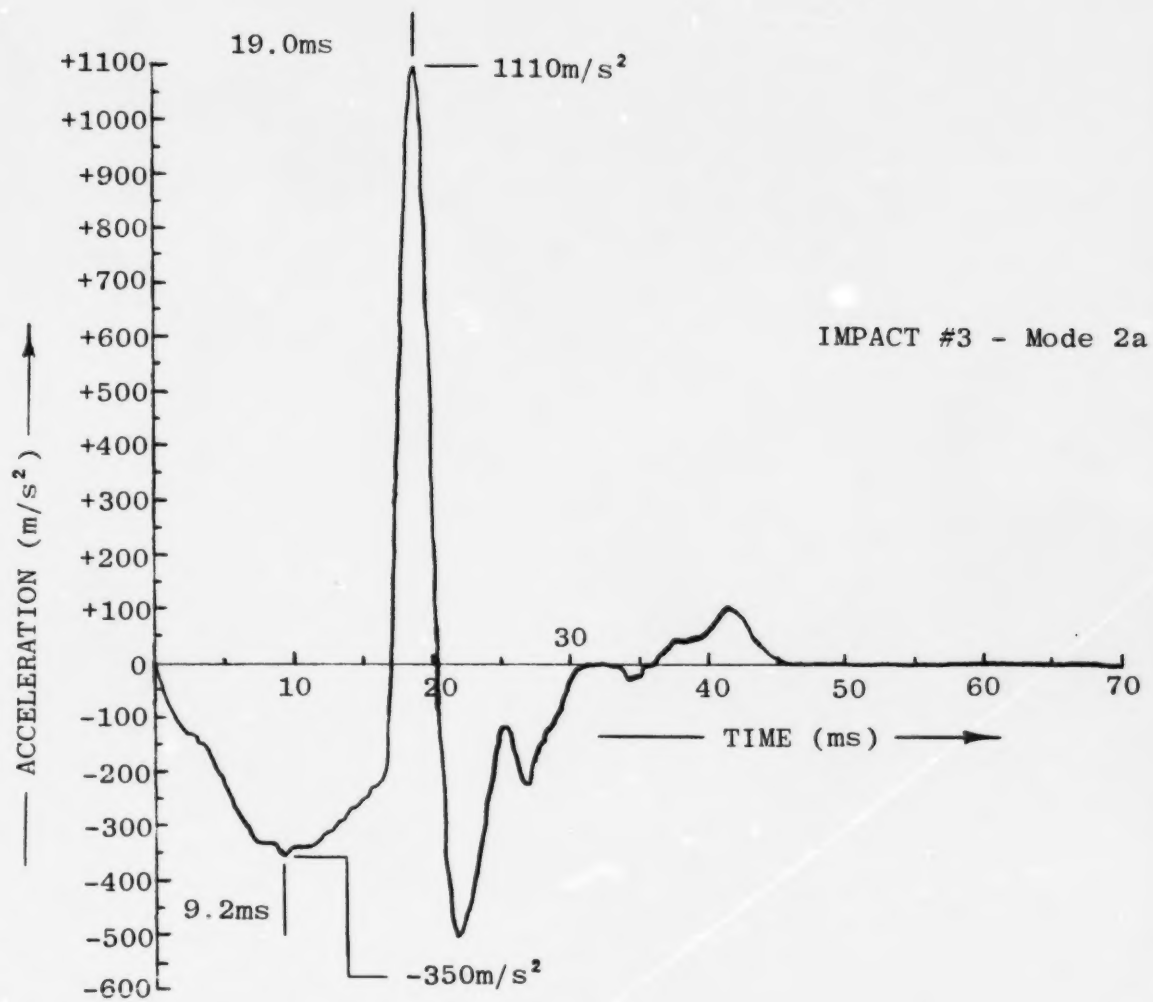


FIGURE 10
TYPICAL TIME HISTORY
FOR AXIAL IMPACT

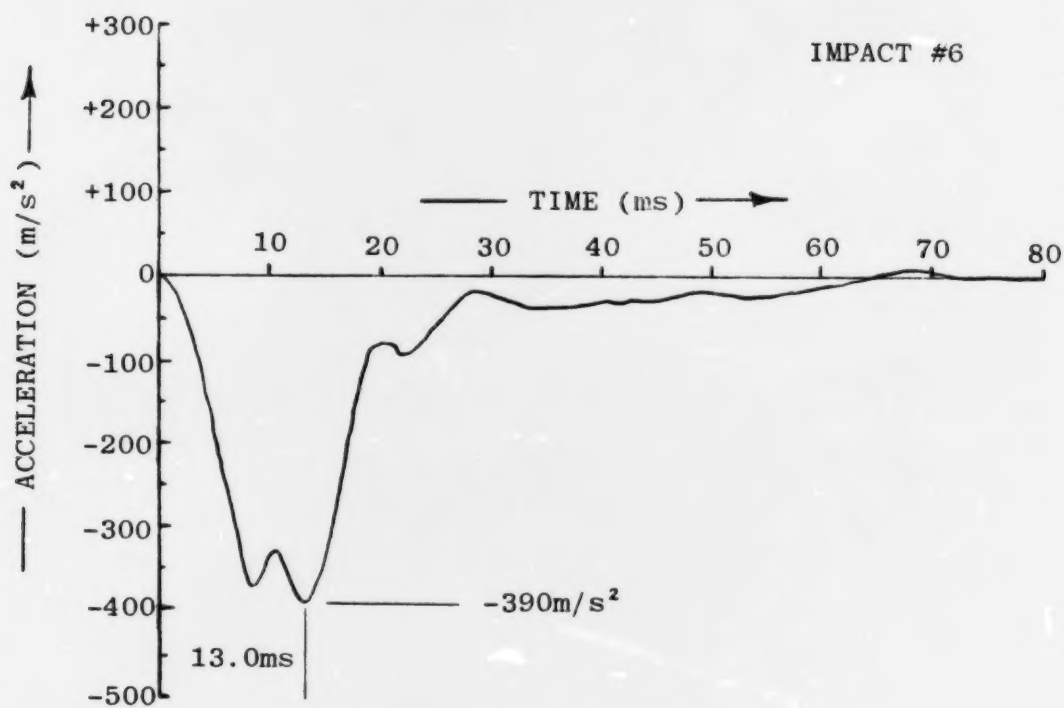


FIGURE 11

TYPICAL TIME HISTORY FOR LATERAL IMPACT

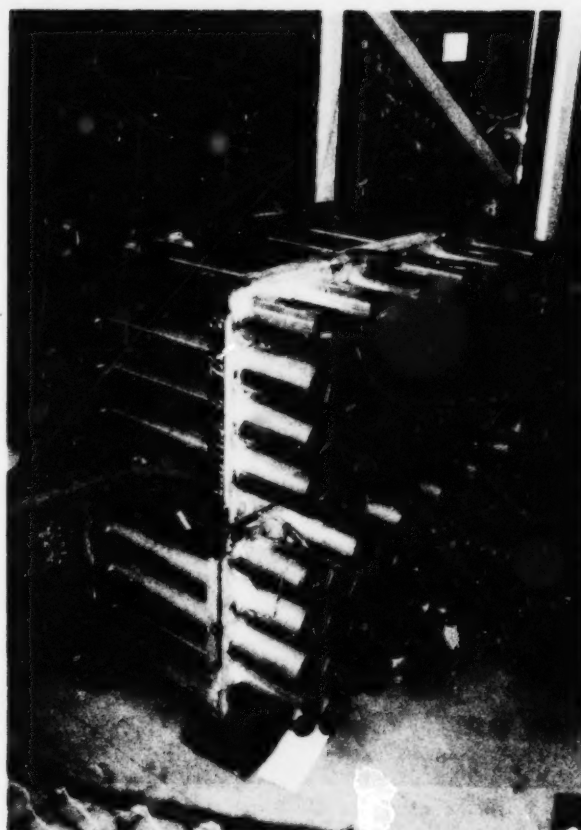
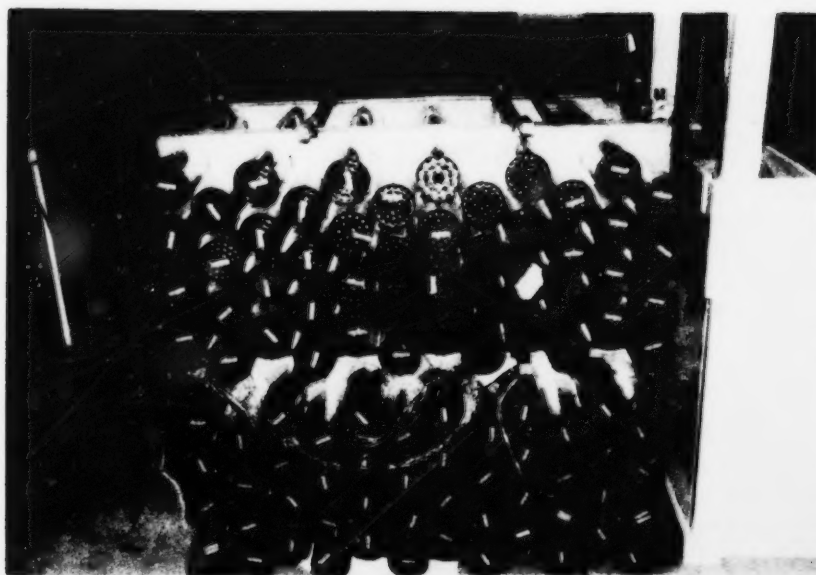


FIGURE 12
TWO MODULE STACK

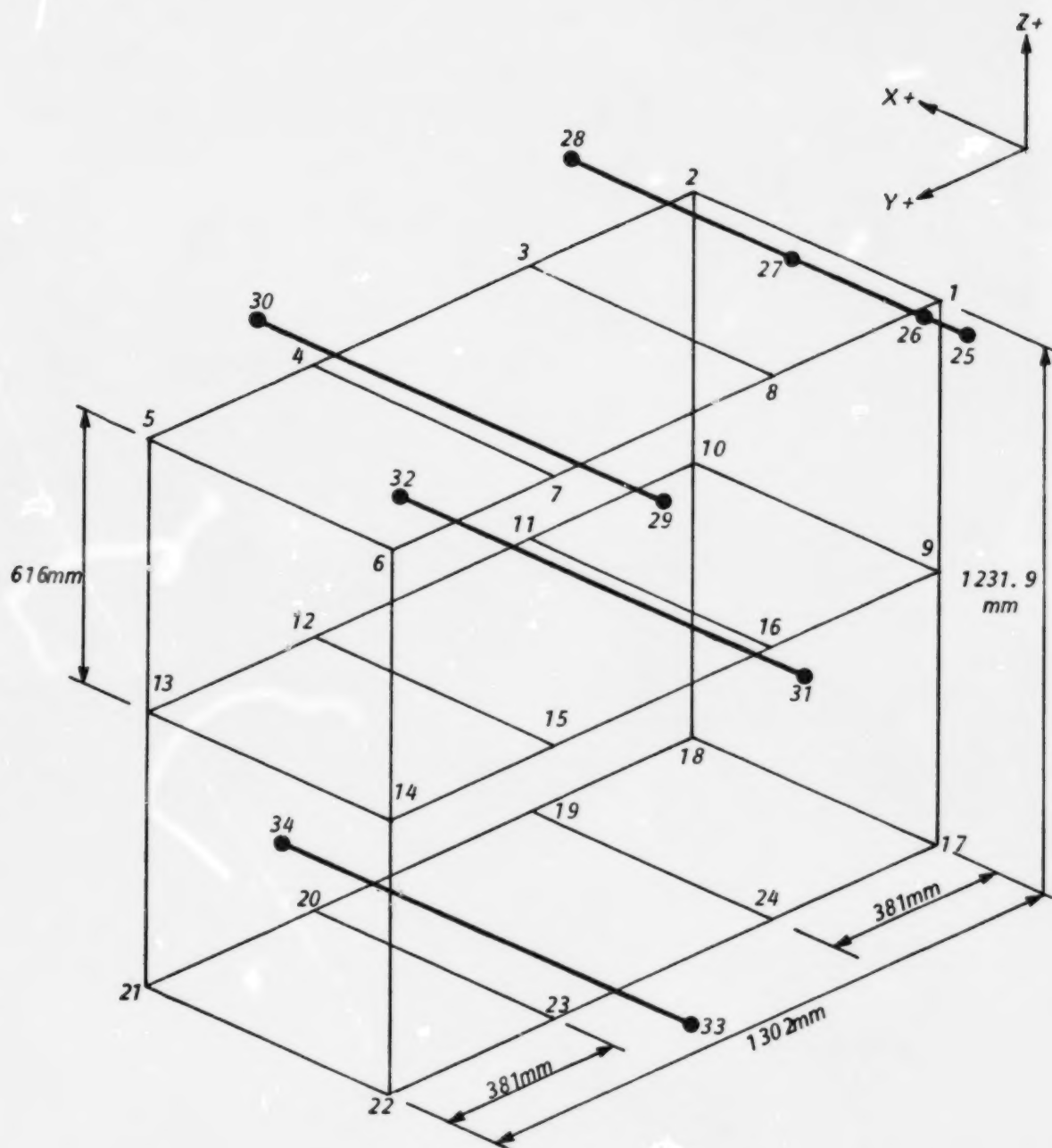


FIGURE 13
MODULE STACK MODEL

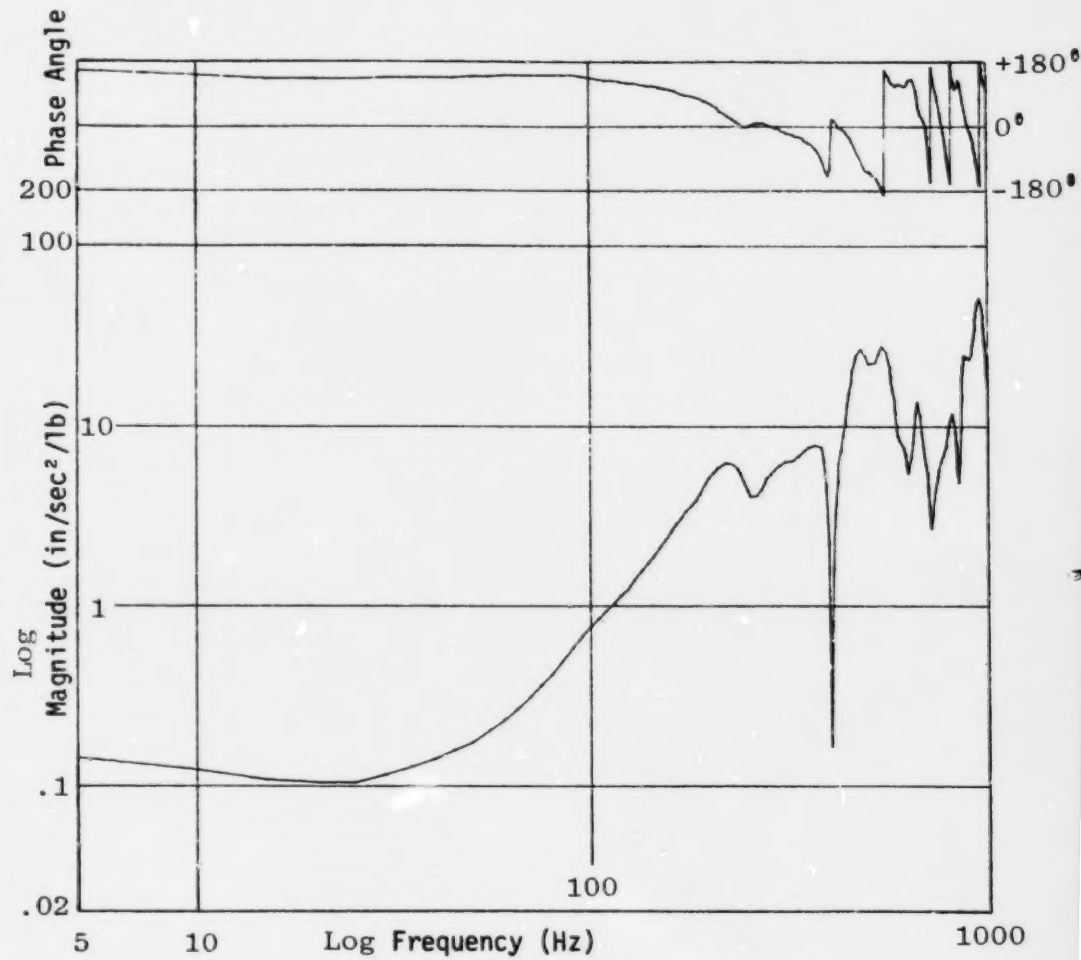


FIGURE 14
MEASURED TRANSFER FUNCTION
X-DIRECTION (HORIZONTAL-AXIAL)

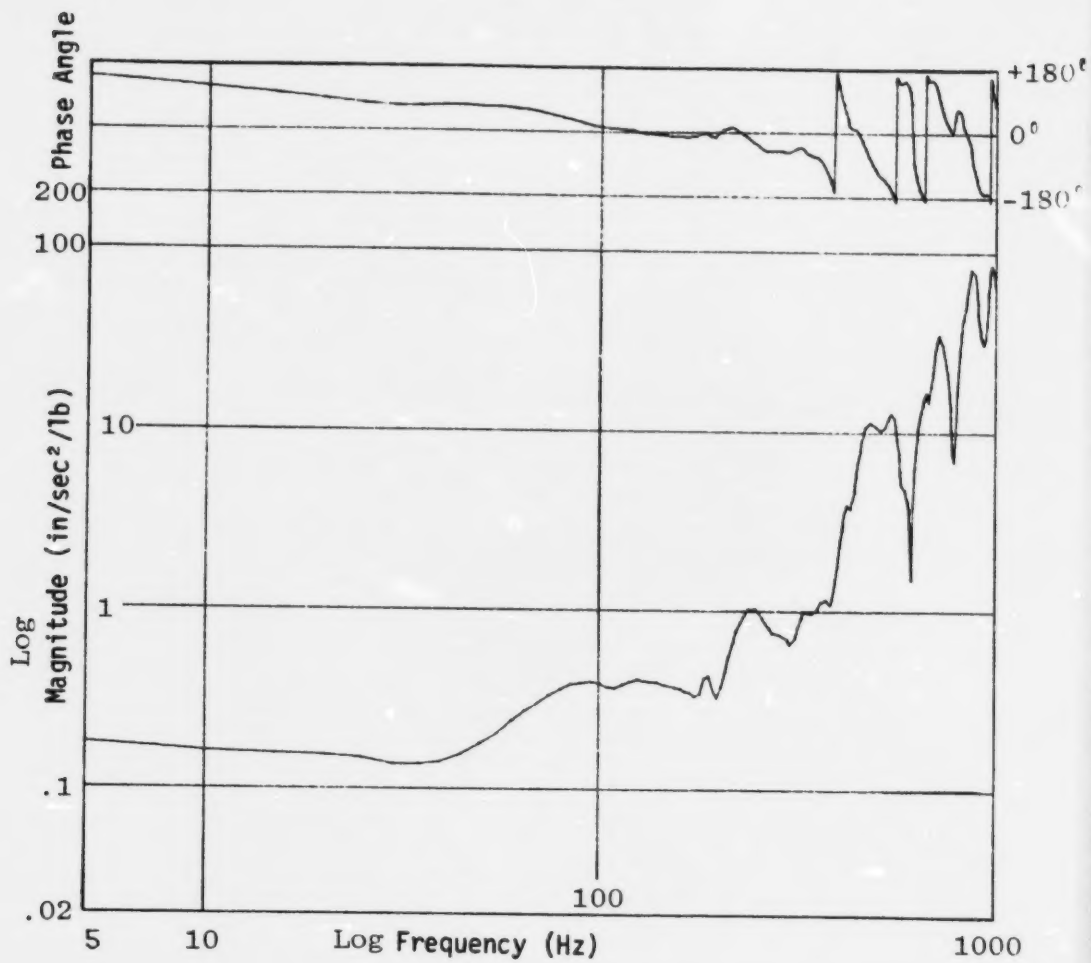


FIGURE 15
MEASURED TRANSFER FUNCTION

Y-DIRECTION (HORIZONTAL-LATERAL)

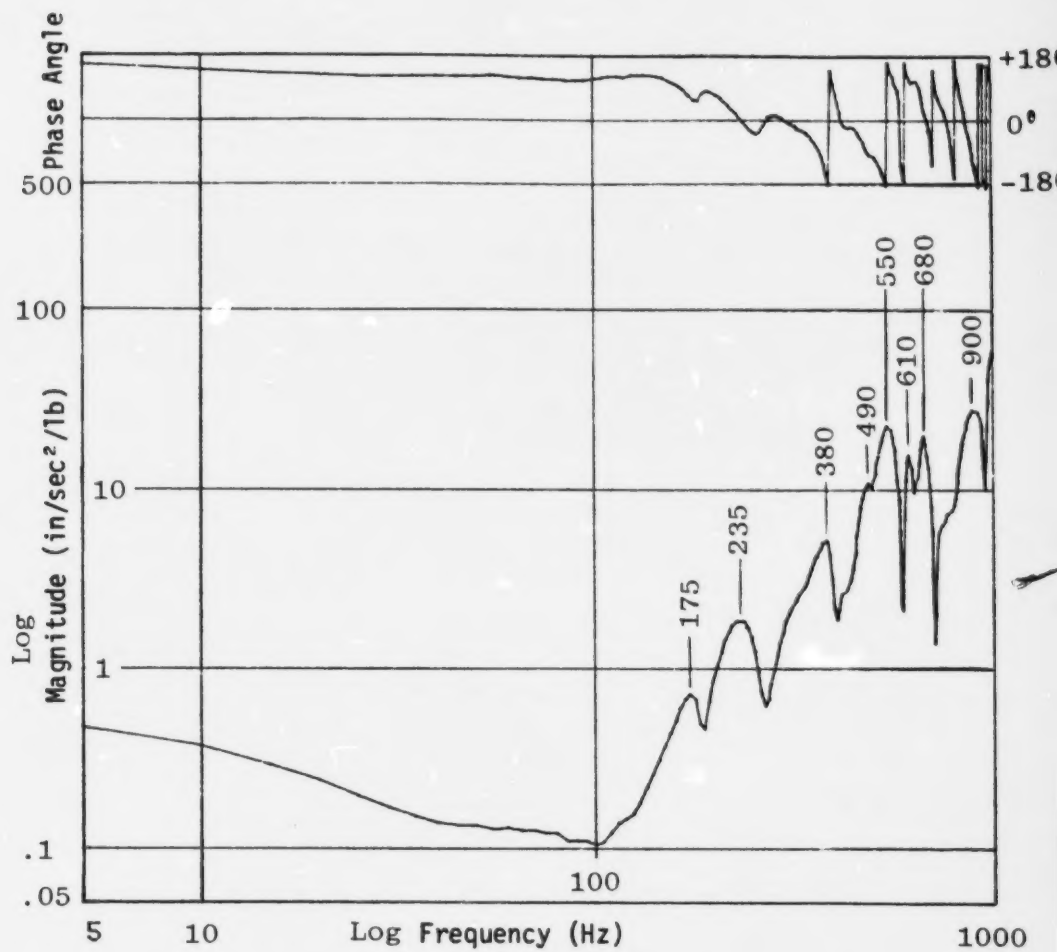
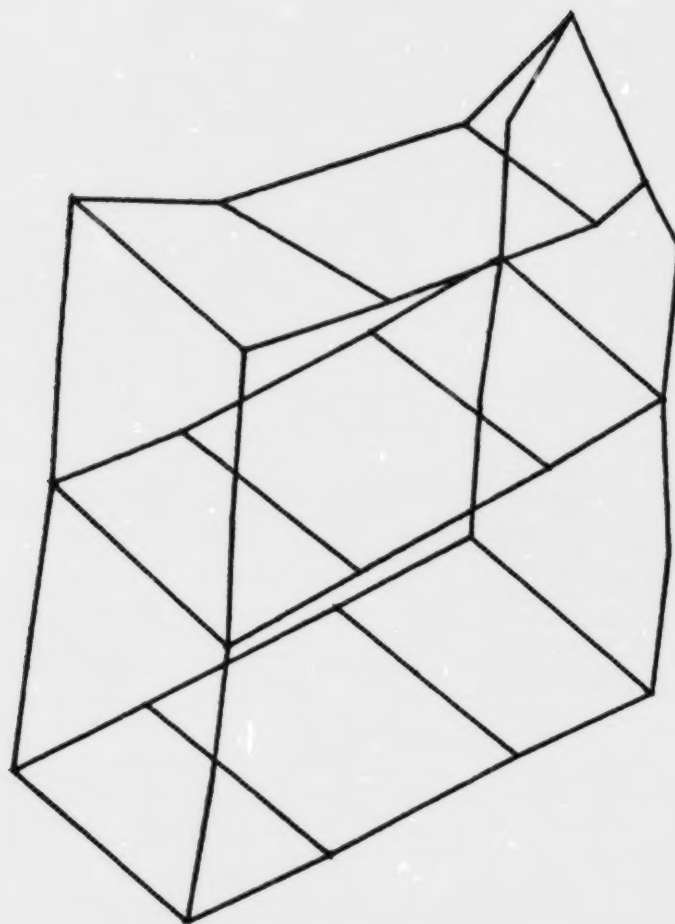


FIGURE 16
MEASURED TRANSFER FUNCTION

Z-DIRECTION (VERTICAL)



F= 196.791 HZ

FIGURE 17
MODE SHAPE NUMBER 1

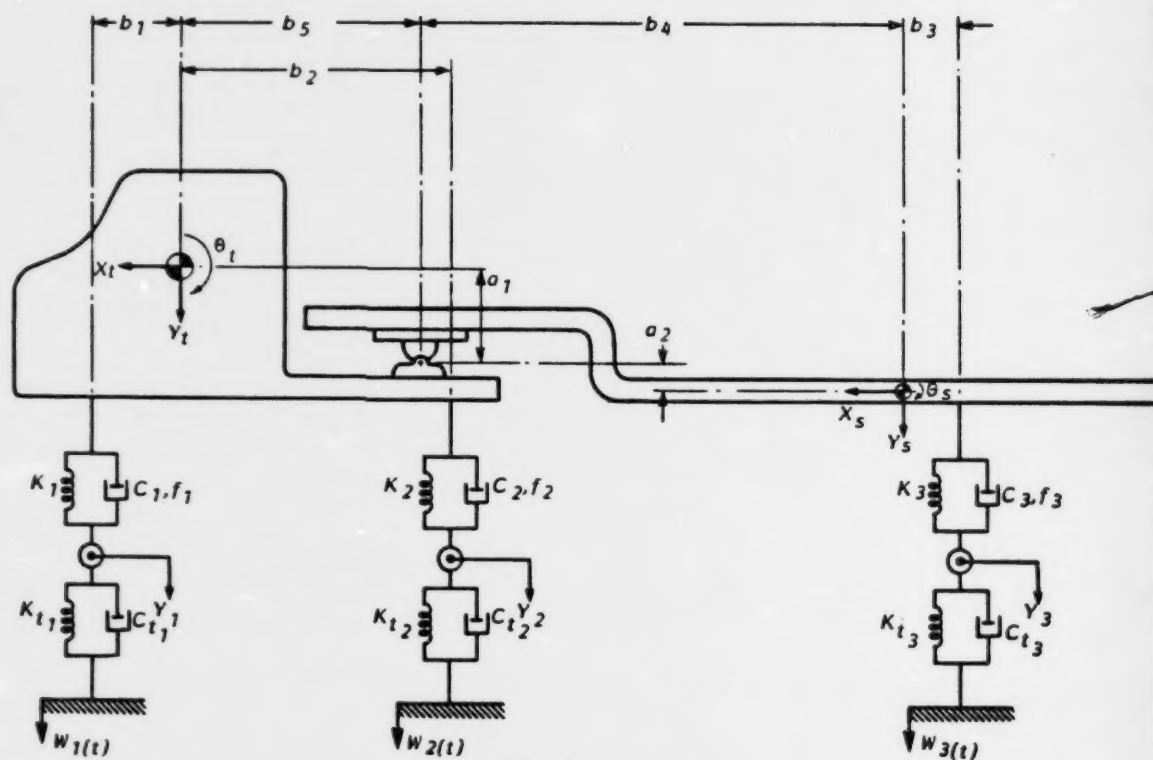


FIGURE 18
THE TRACTOR - SEMITRAILER MODEL

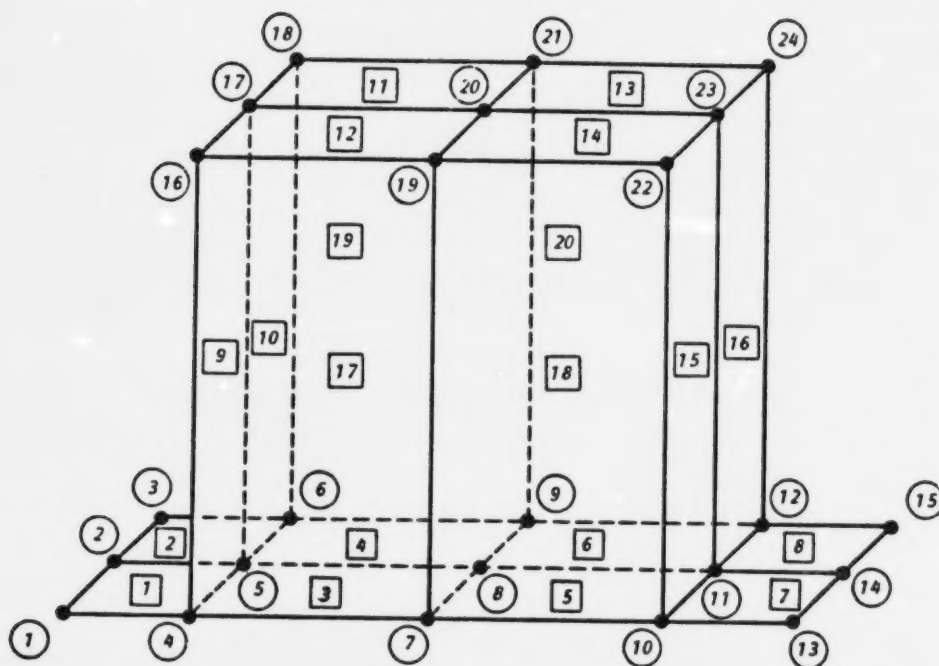


FIGURE 19

FINITE ELEMENT MODEL FOR CASK/TIEDOWN/TRANSPORT FRAME SYSTEM

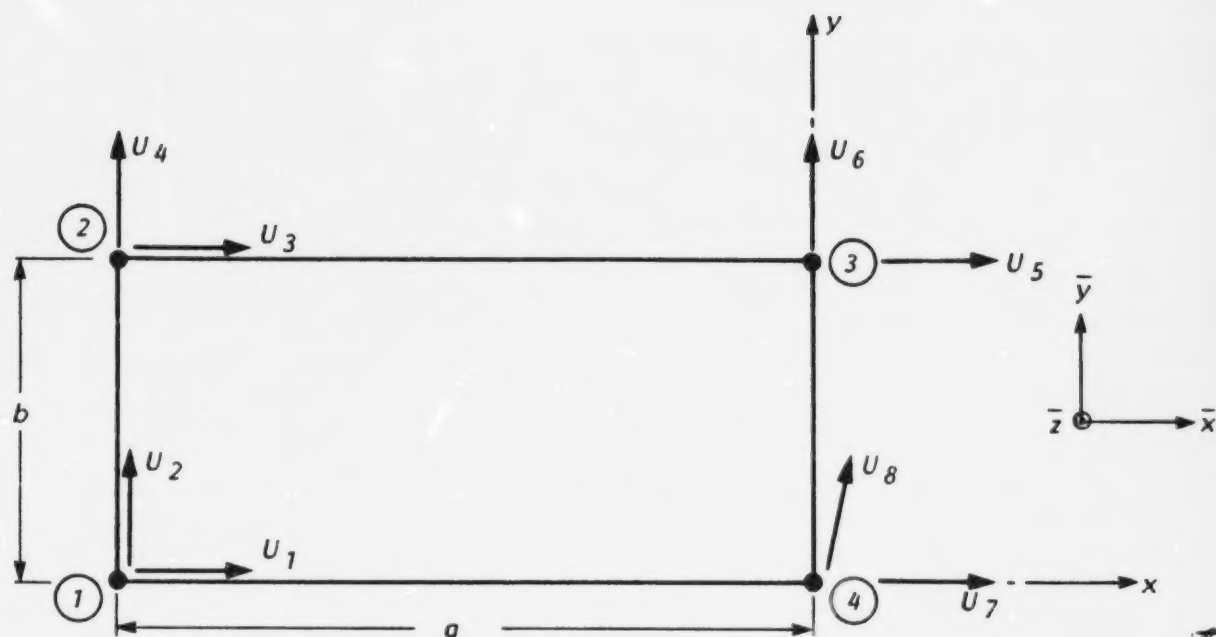


FIGURE 20
RECTANGULAR PLATE ELEMENT (IN. PLANE FORCES)

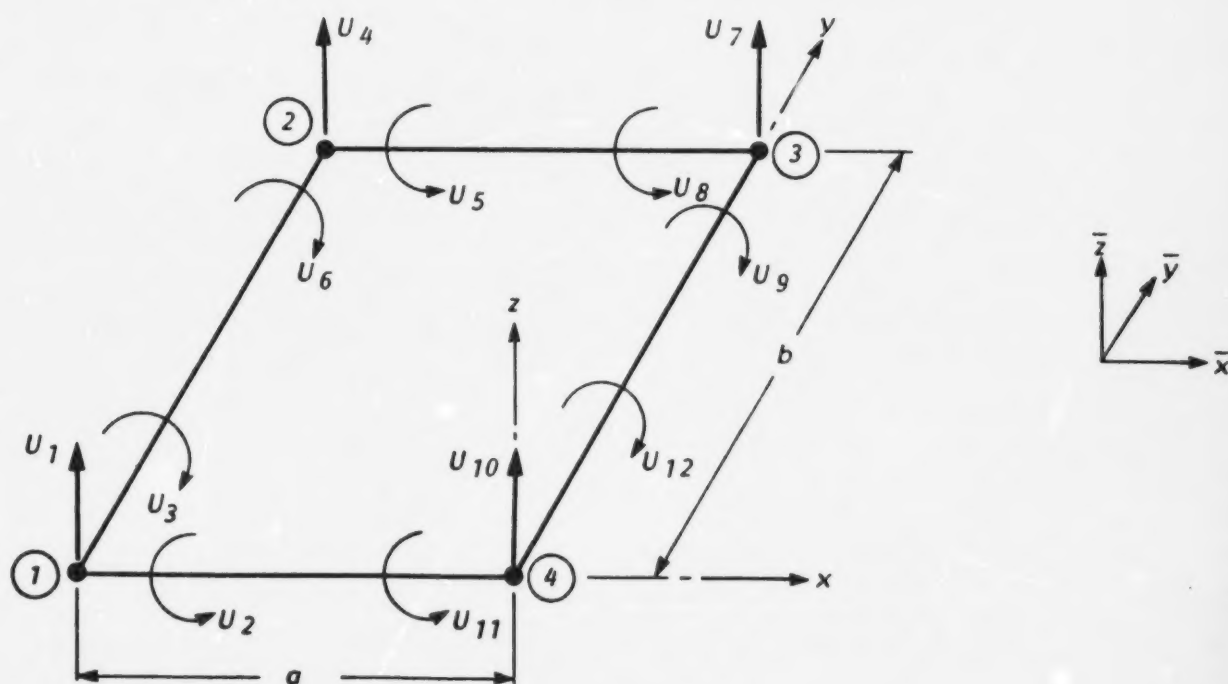


FIGURE 21
RECTANGULAR PLATE ELEMENT (BENDING FORCES)

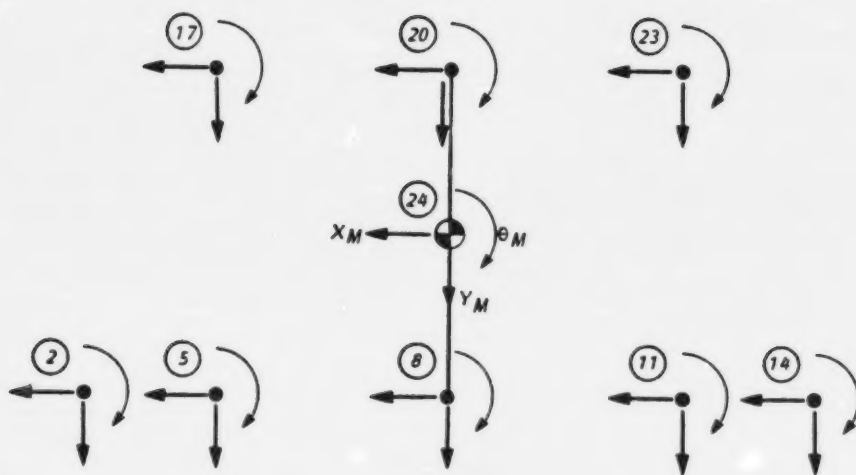


FIGURE 22
REDUCED MODEL FOR CASK/TIEDOWN/TRANSPORT FRAME AND MODULE SYSTEM

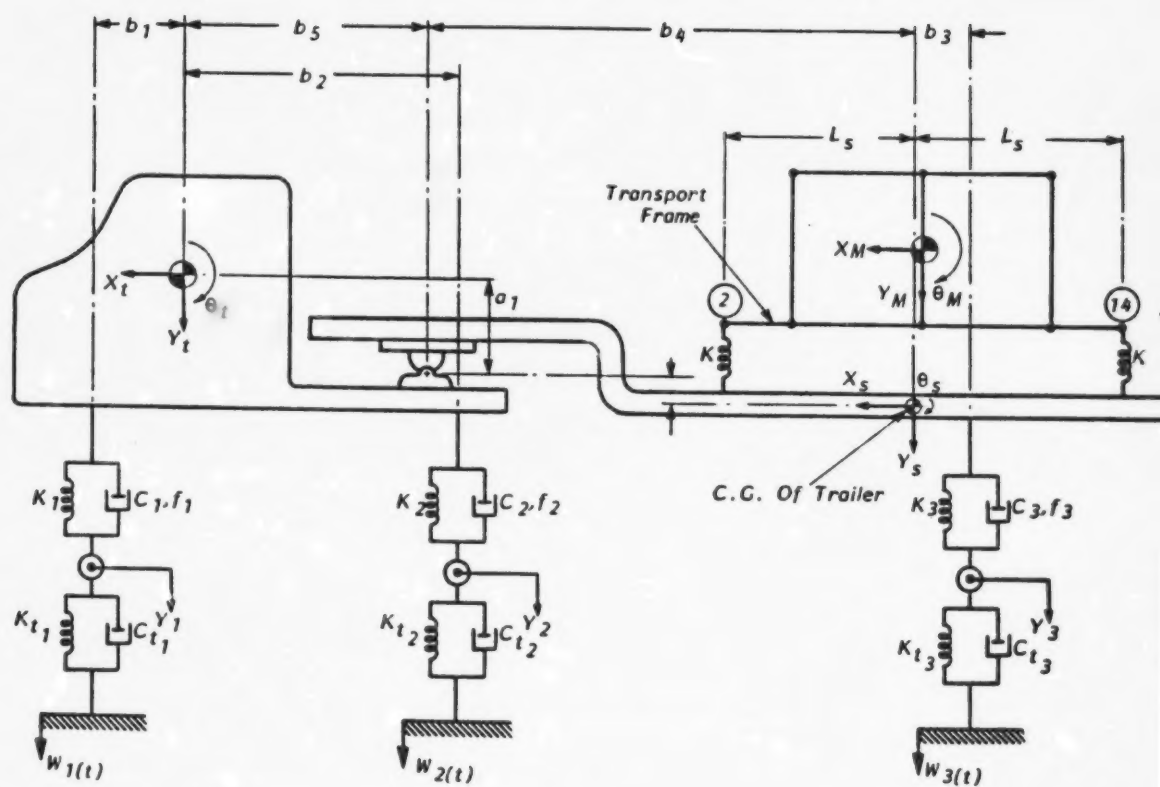


FIGURE 23
ARTICULATED VEHICLE MODEL

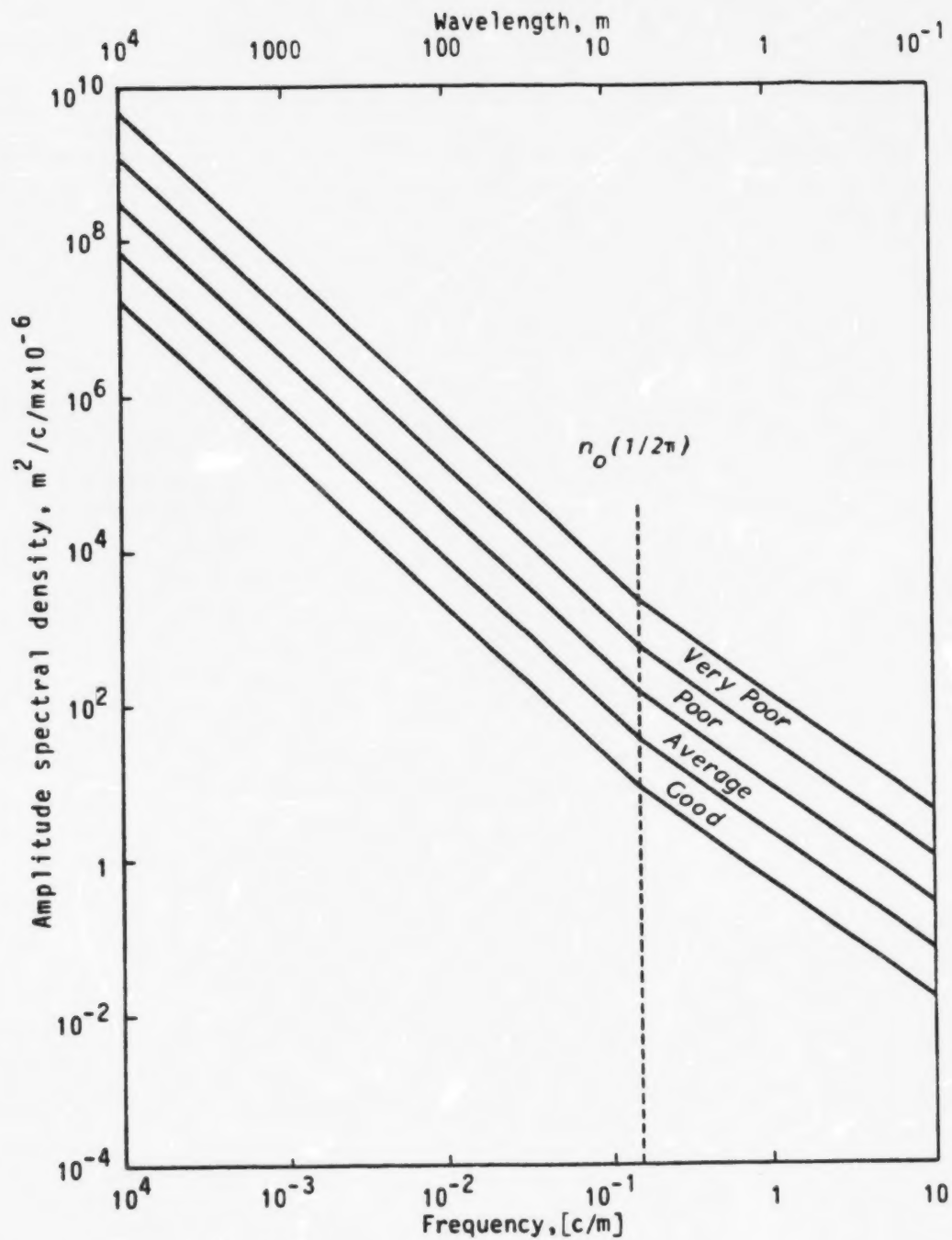


FIGURE 24
CLASSIFICATION OF ROAD SURFACES BY POWER SPECTRAL DENSITY

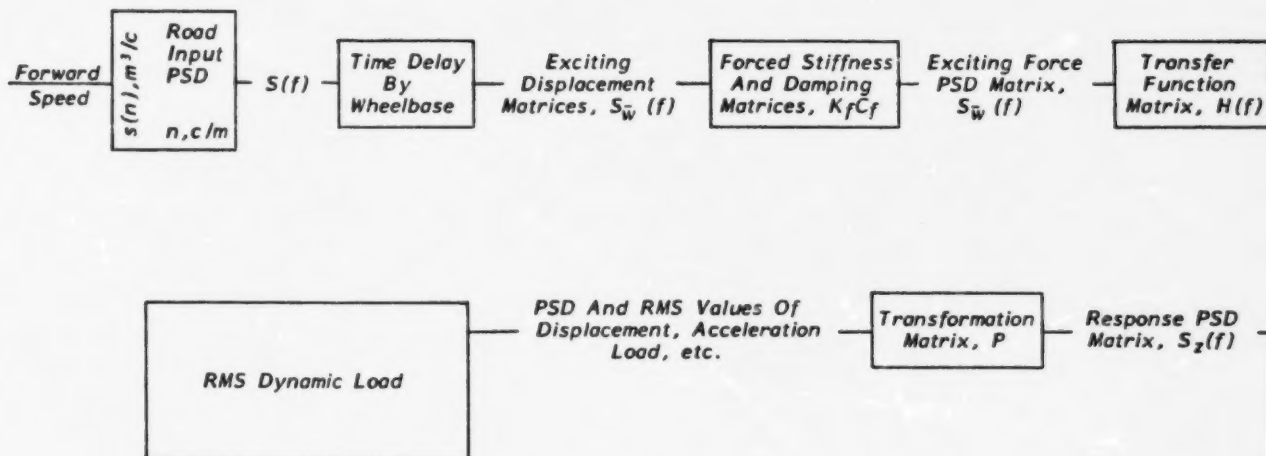


FIGURE 25
VEHICLE RIDE ANALYSIS

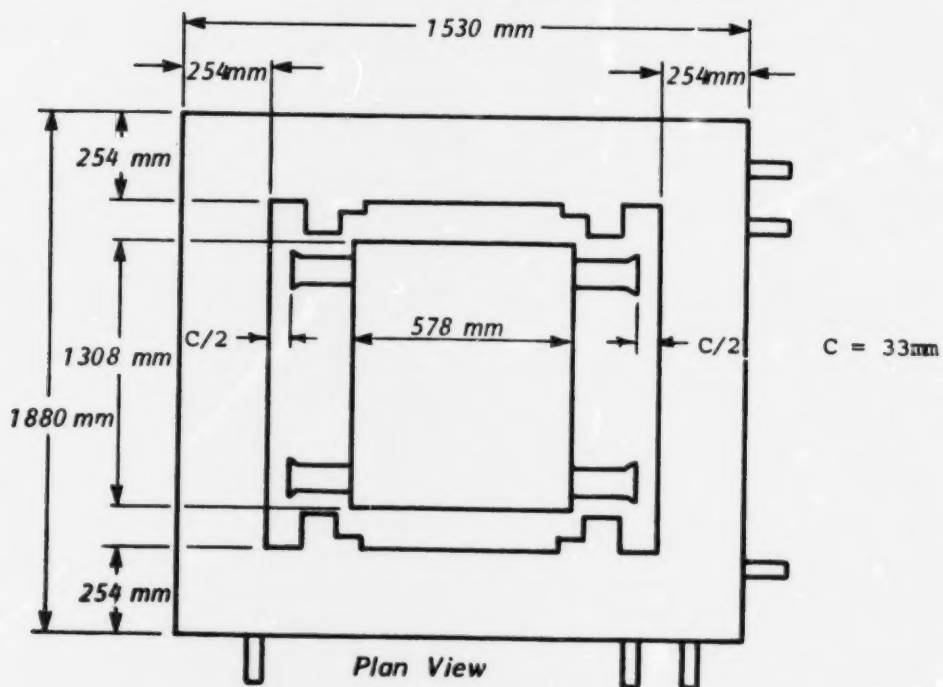
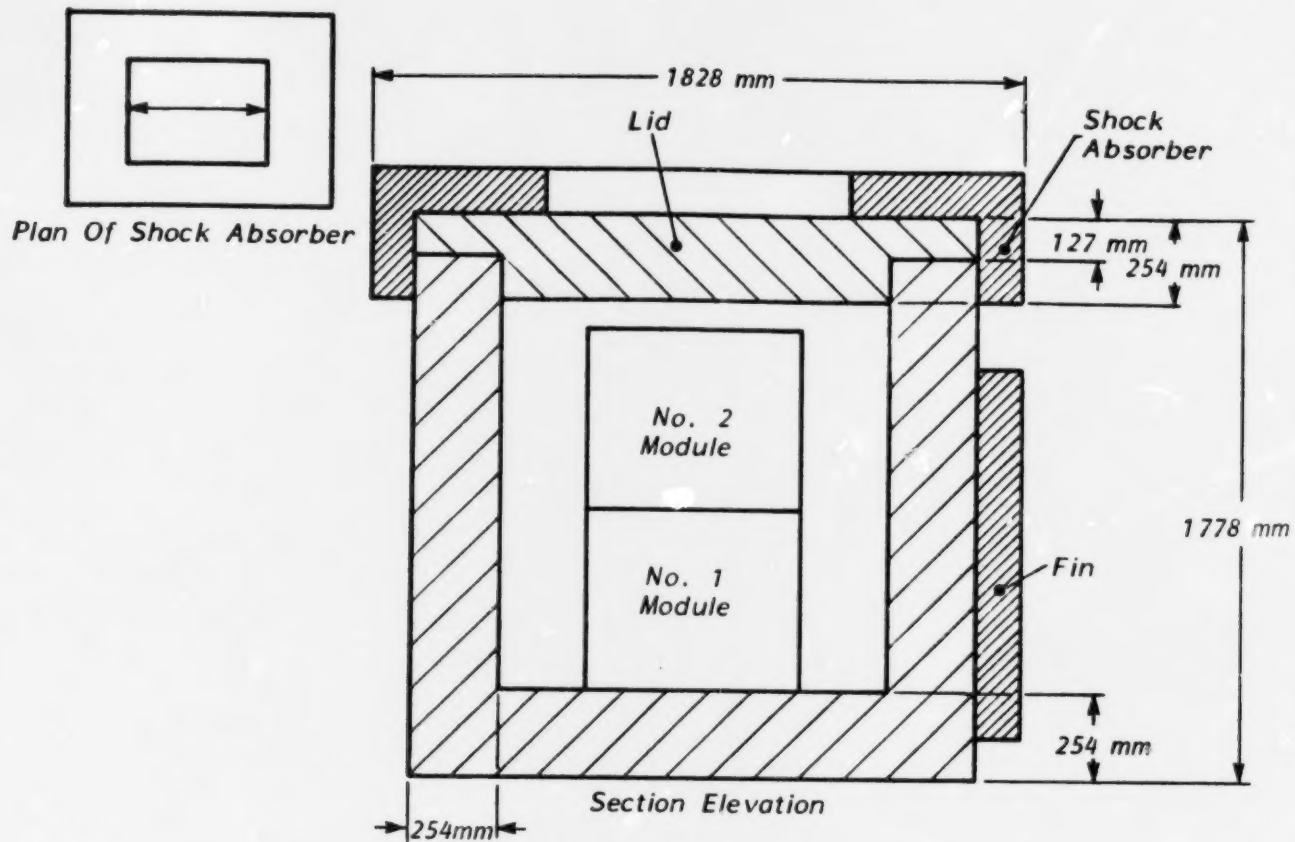


FIGURE 26
ROAD CASK

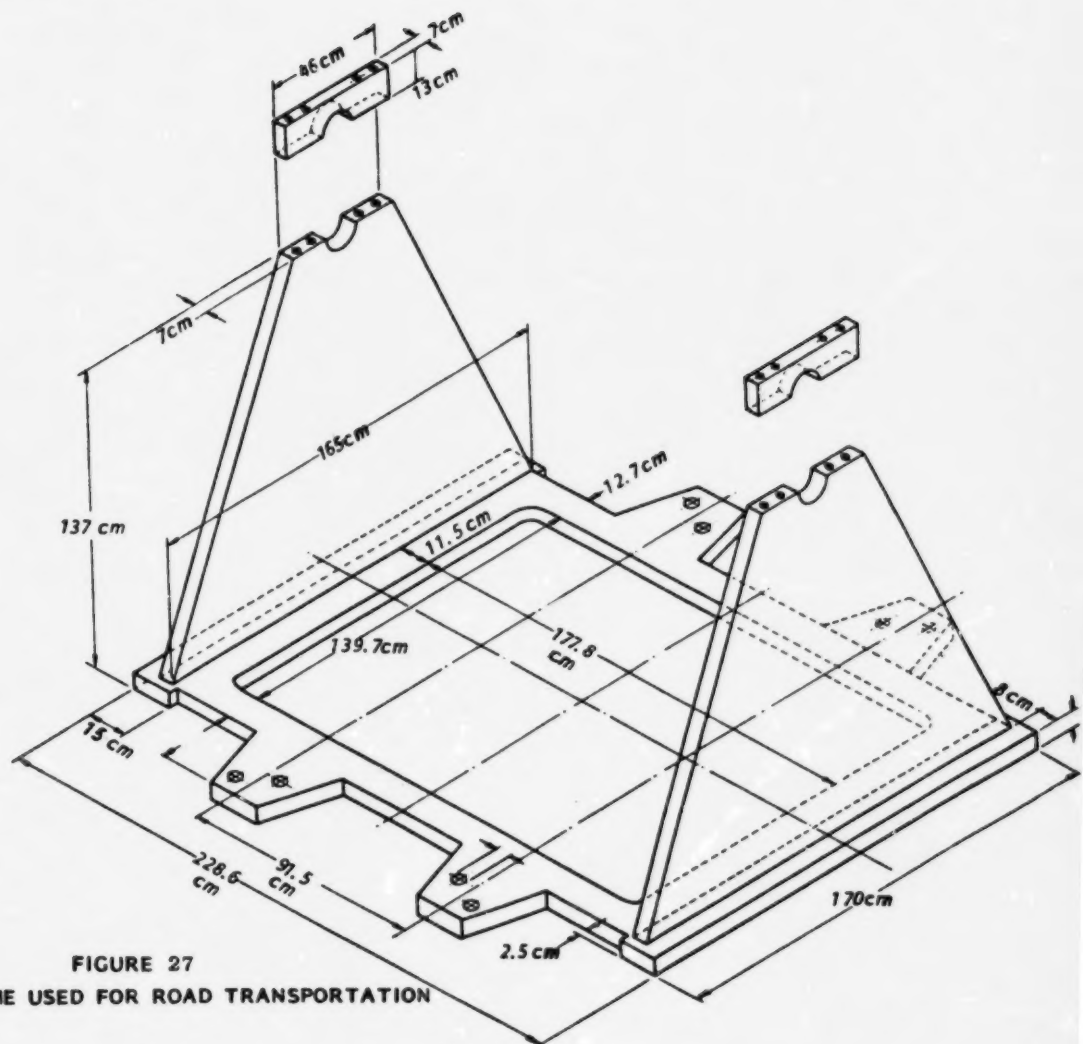


FIGURE 27
SUPPORT FRAME USED FOR ROAD TRANSPORTATION

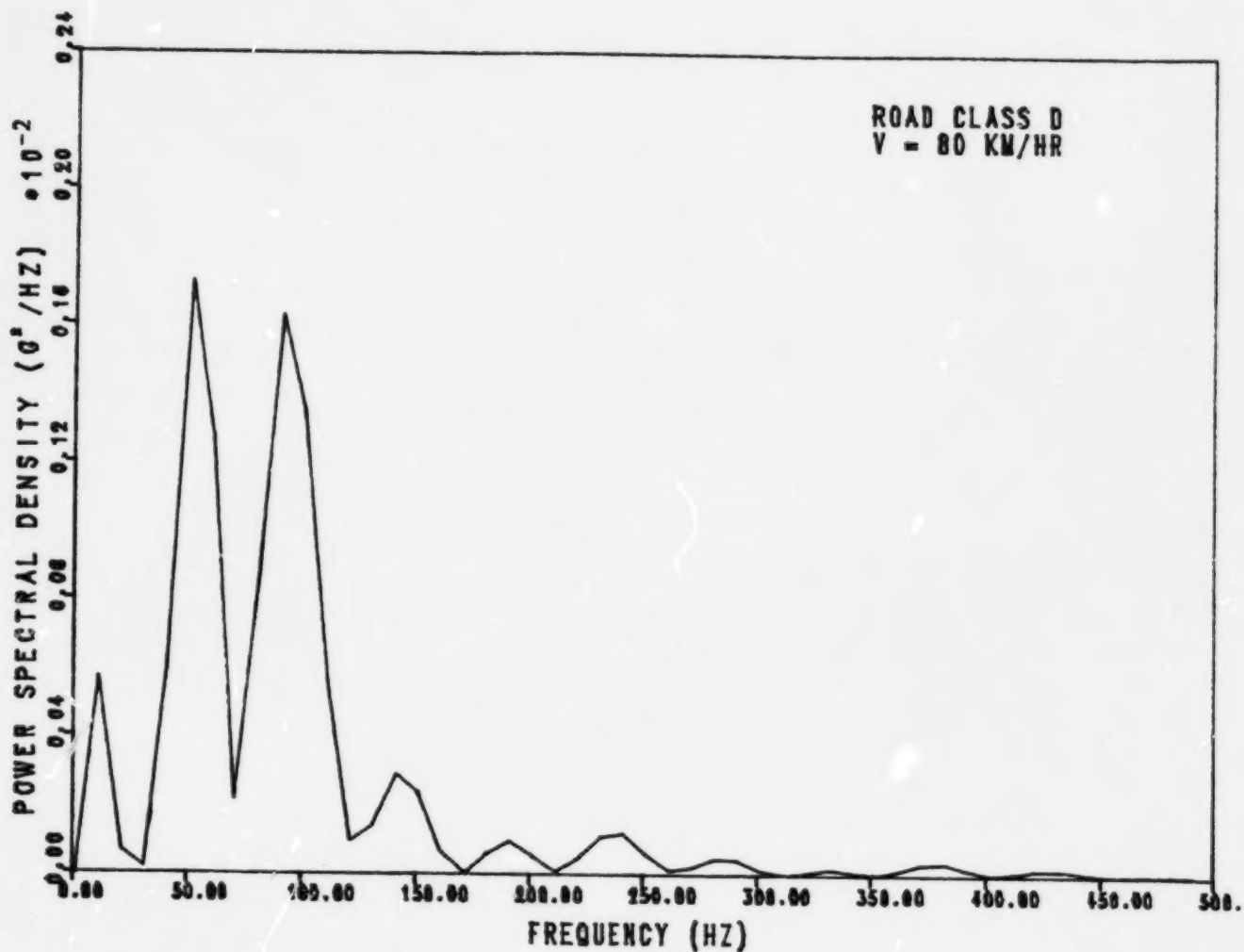


FIGURE 28

THE POWER SPECTRAL DENSITY OF THE VERTICAL ACCELERATION
OF THE MODULE STACK CENTRE OF GRAVITY VERSUS
FREQUENCY FOR ROAD CLASS D

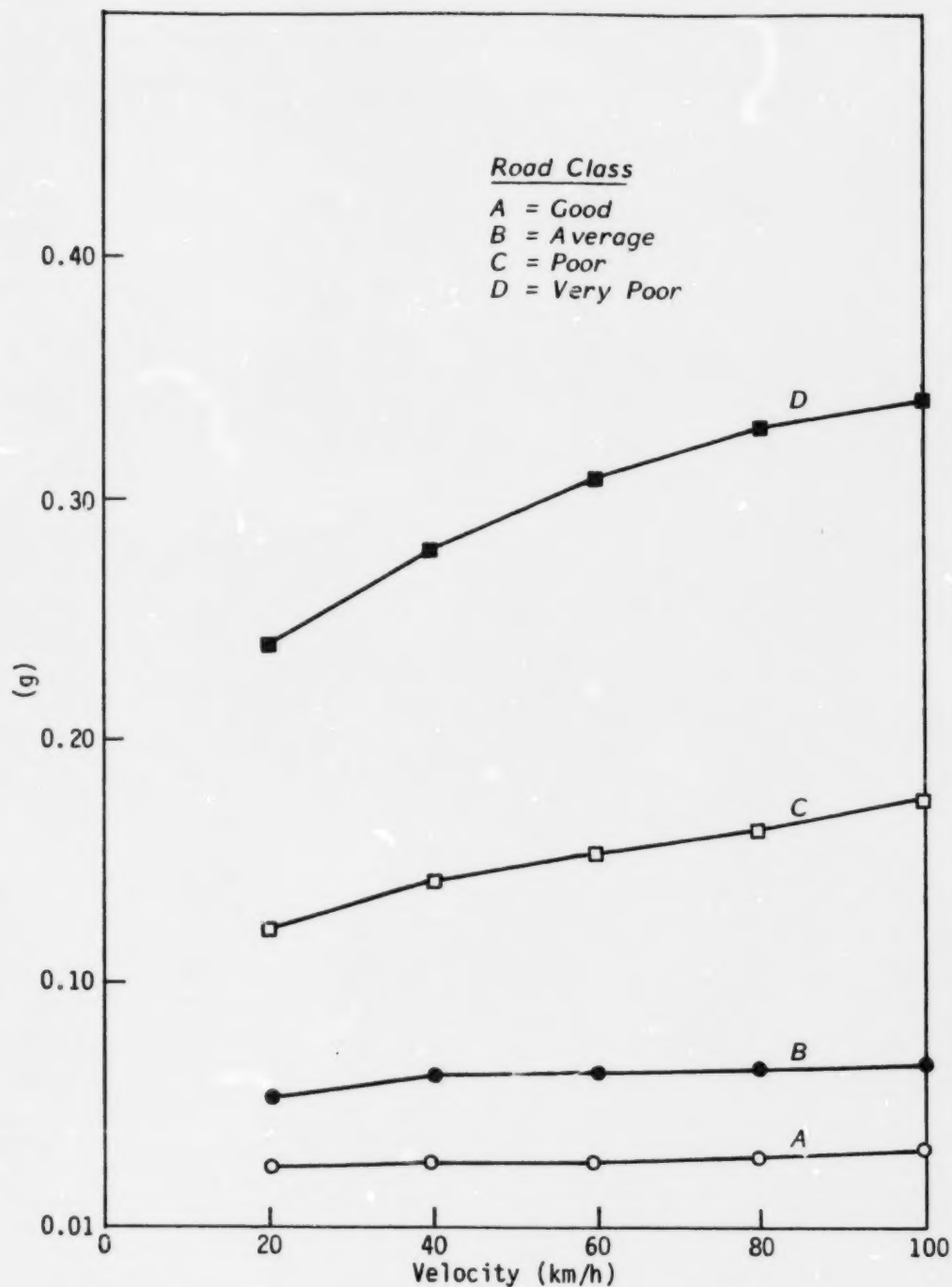


FIGURE 29
R.M.S. OF VERTICAL ACCELERATION OF THE MODULE STACK C.G.
VERSUS FORWARD VELOCITY

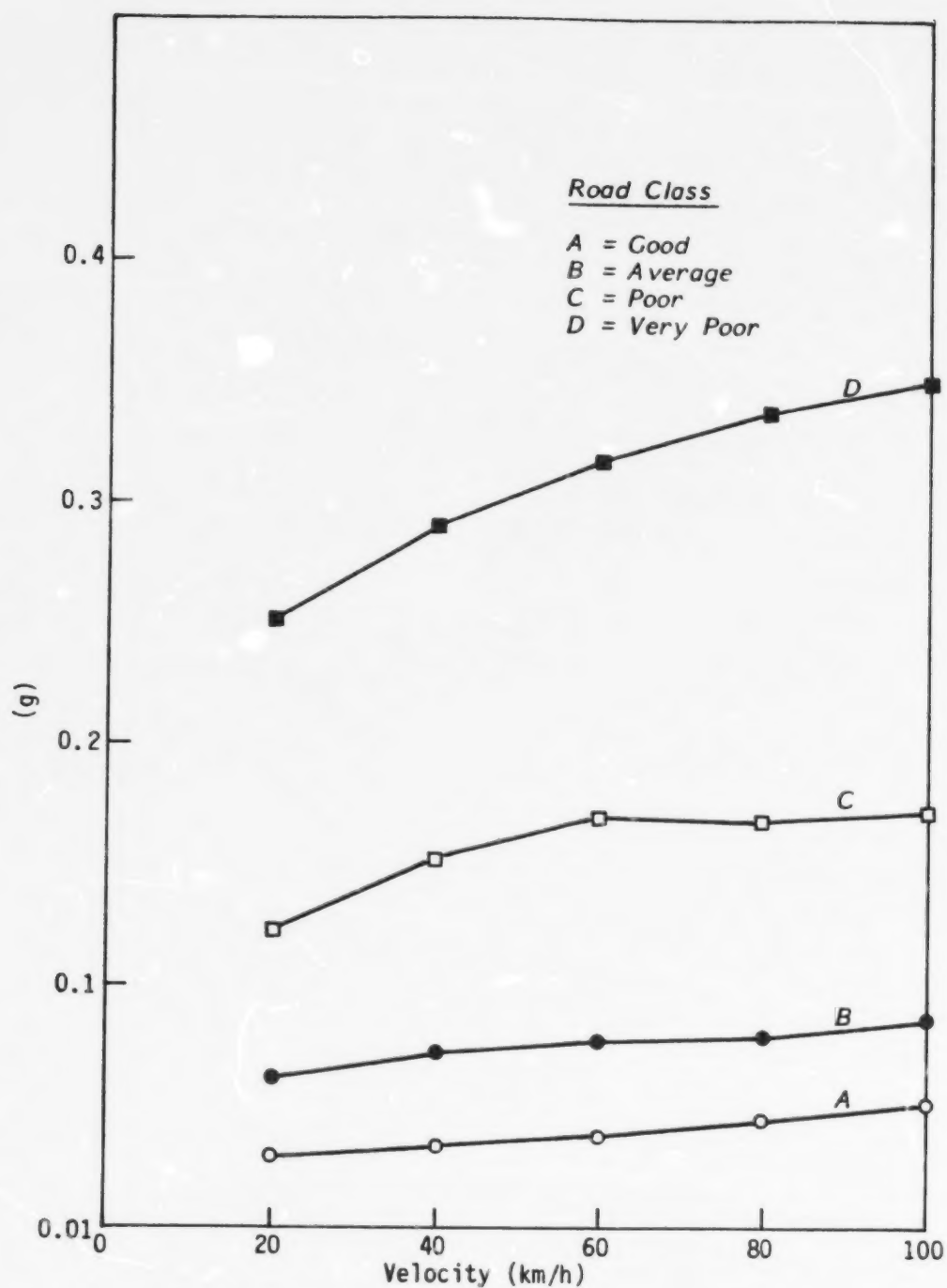


FIGURE 30
R.M.S. OF VERTICAL ACCELERATION AT TIE DOWN
CONNECTION WITH THE SEMI TRAILER

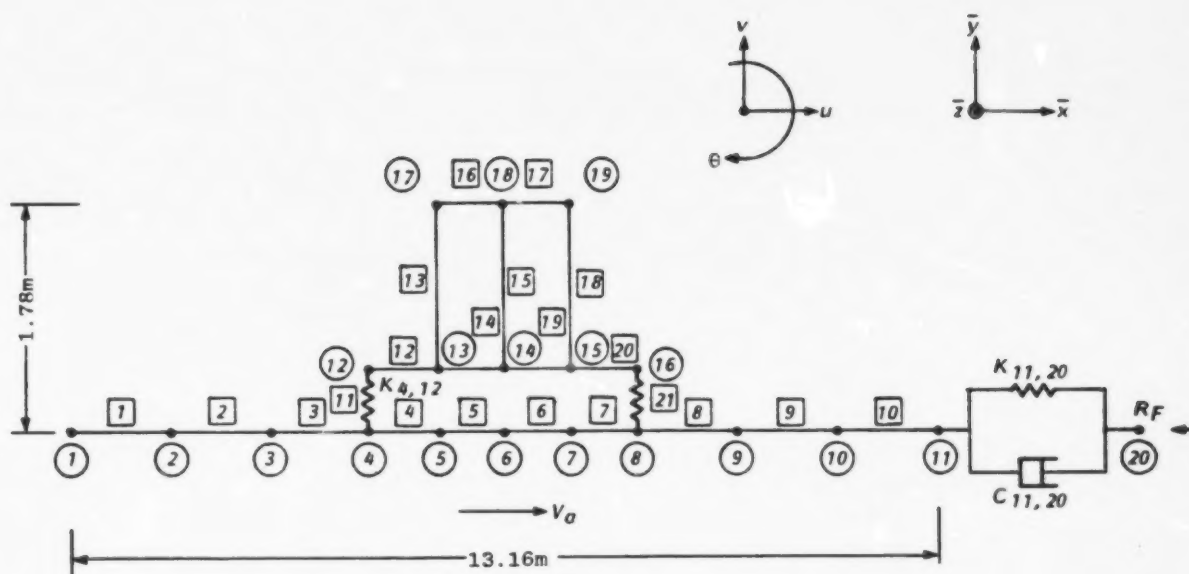


FIGURE 31
STRUCTURAL MODELING TECHNIQUE FOR RAIL CAR, CASK, TRANSPORT
FRAME, SHIPPING MODULES, AND COUPLER

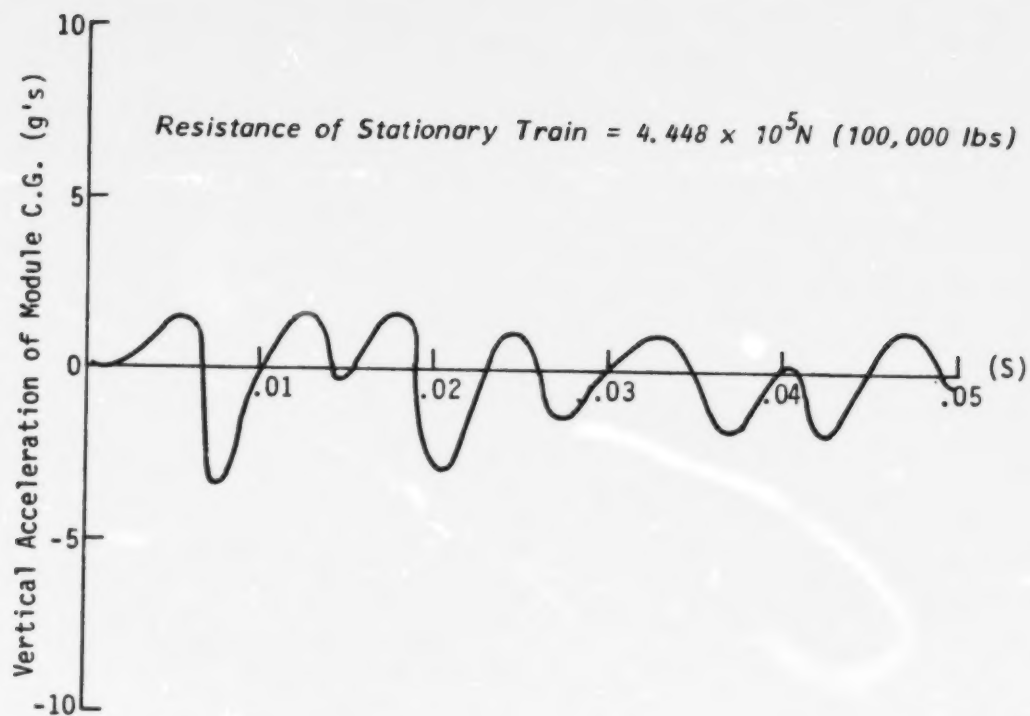


FIGURE 32
ACCELERATION TIME PROFILE FOR RAIL CAR CARRYING 38-TONNE
CARGO STRIKING 226.8-TONNE STATIONARY TRAIN AT 16 km/h

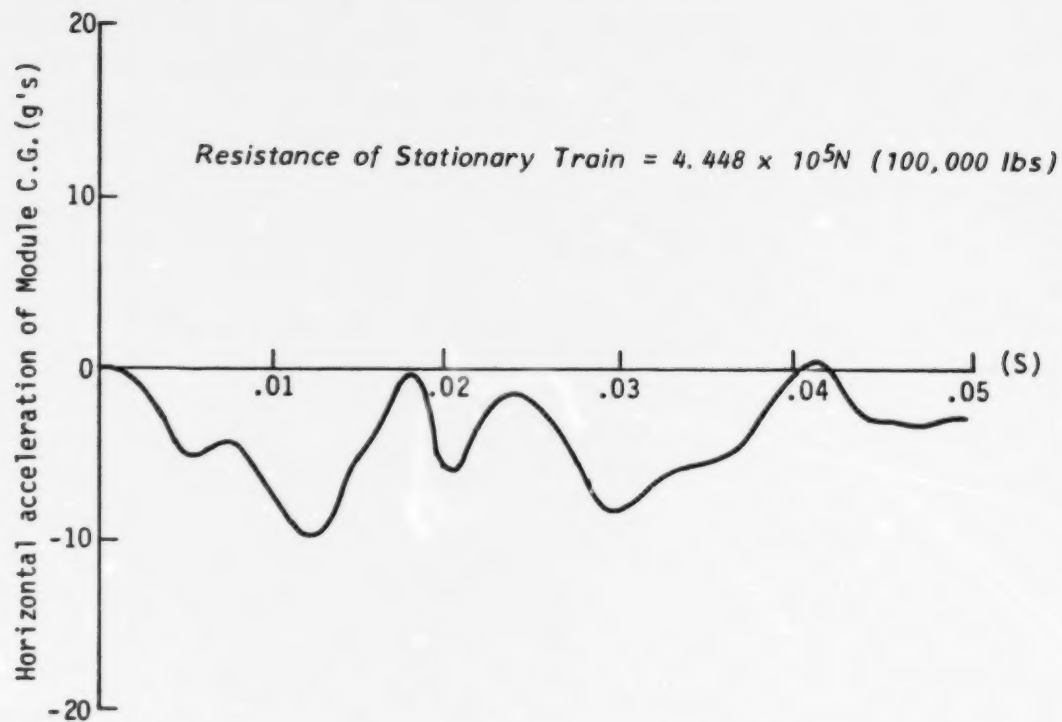


FIGURE 33
ACCELERATION TIME PROFILE FOR RAIL CAR CARRYING
38-TONNE CARGO STRIKING 226.8-TONNE STATIONARY
TRAIN AT 16 km/h

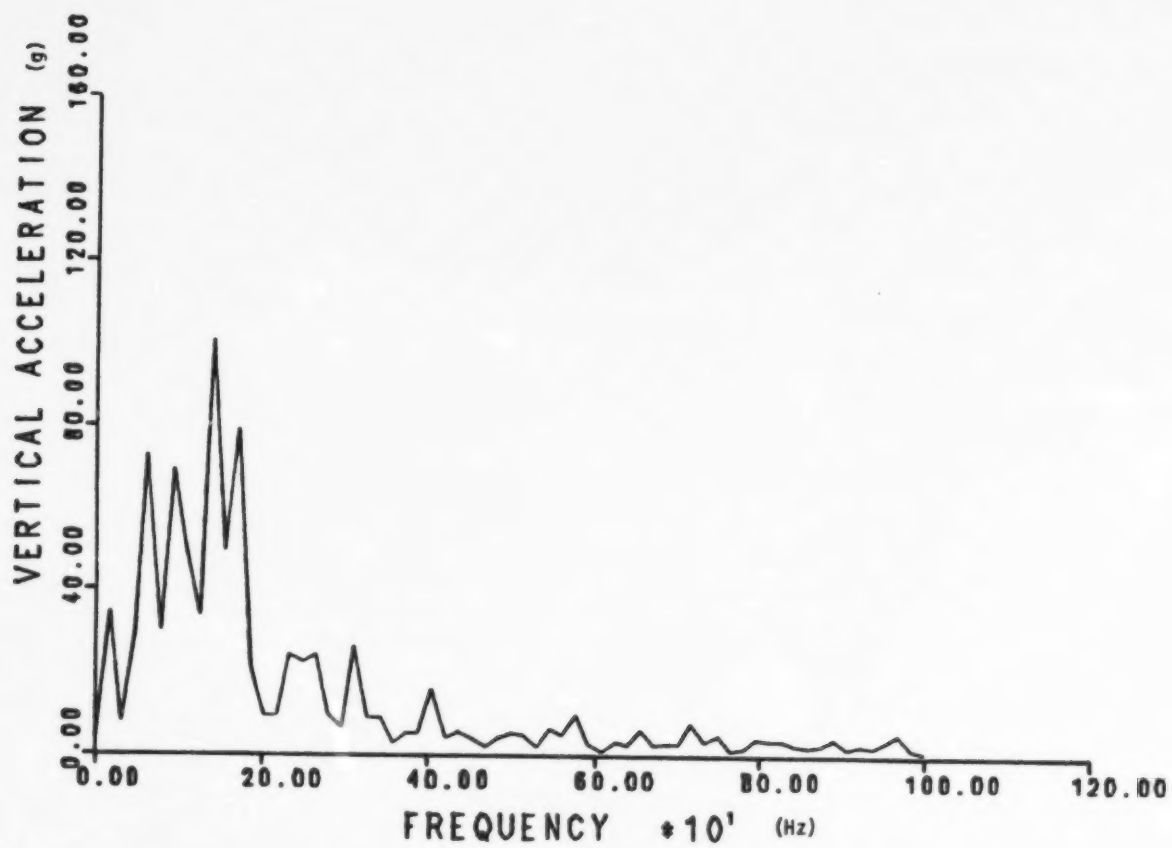


FIGURE 34
VERTICAL ACCELERATION FREQUENCY PROFILE

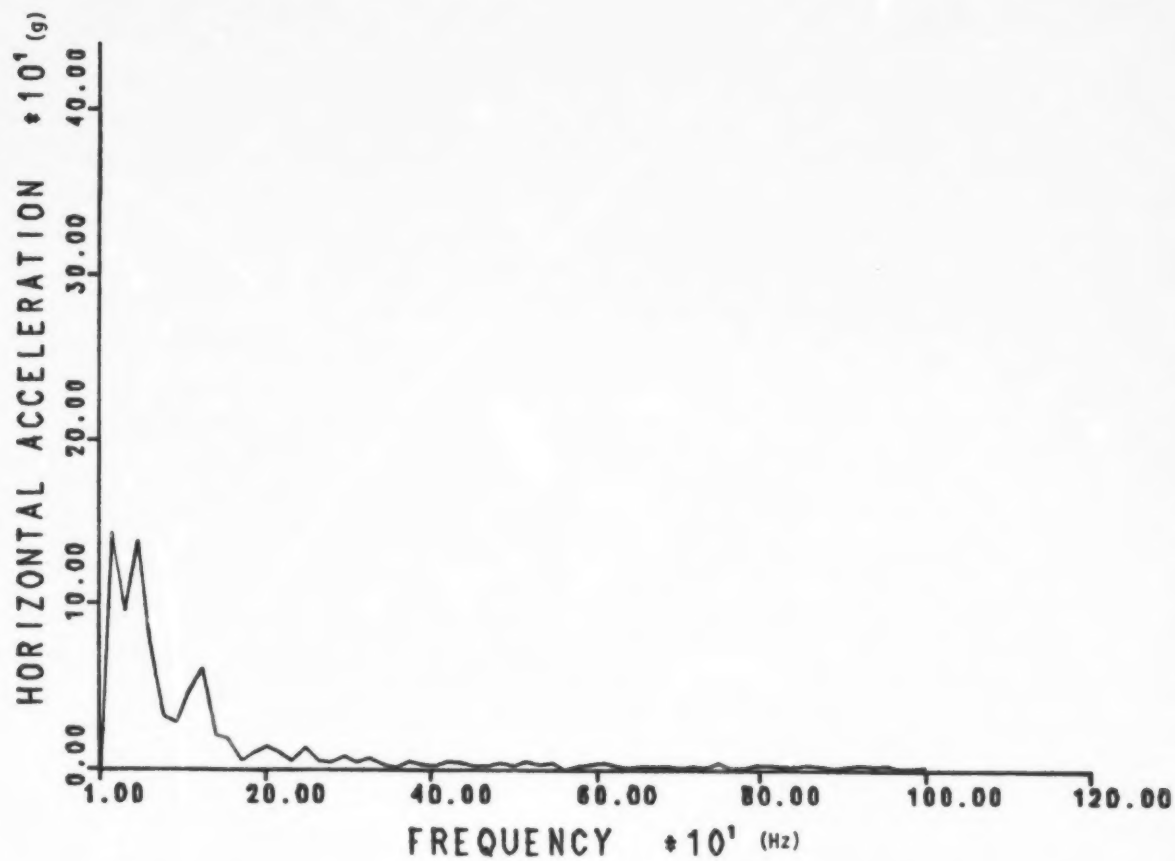


FIGURE 35
HORIZONTAL ACCELERATION FREQUENCY PROFILE

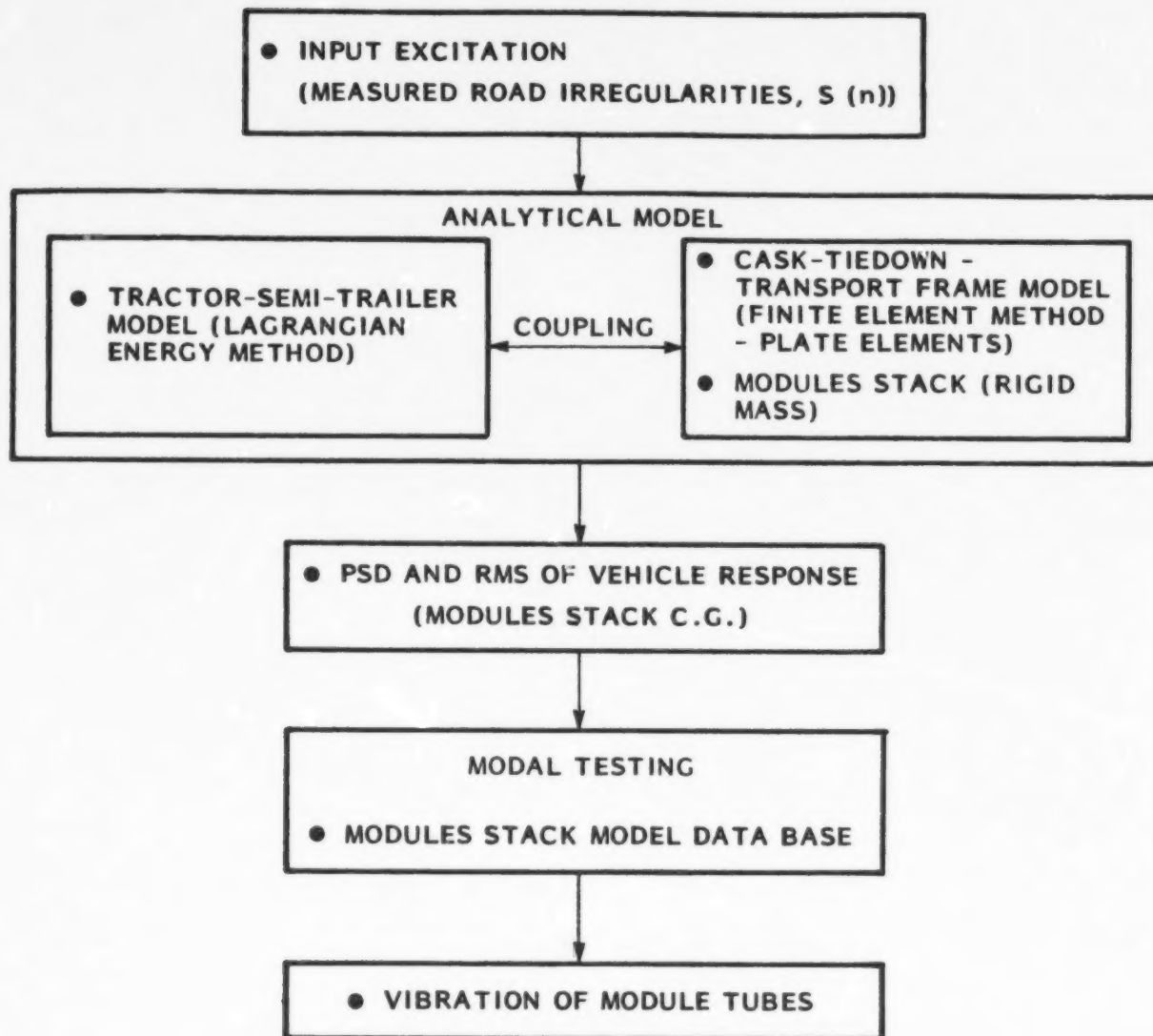


FIGURE 36
MODULE ANALYSIS: ROAD TRANSPORTATION MODEL

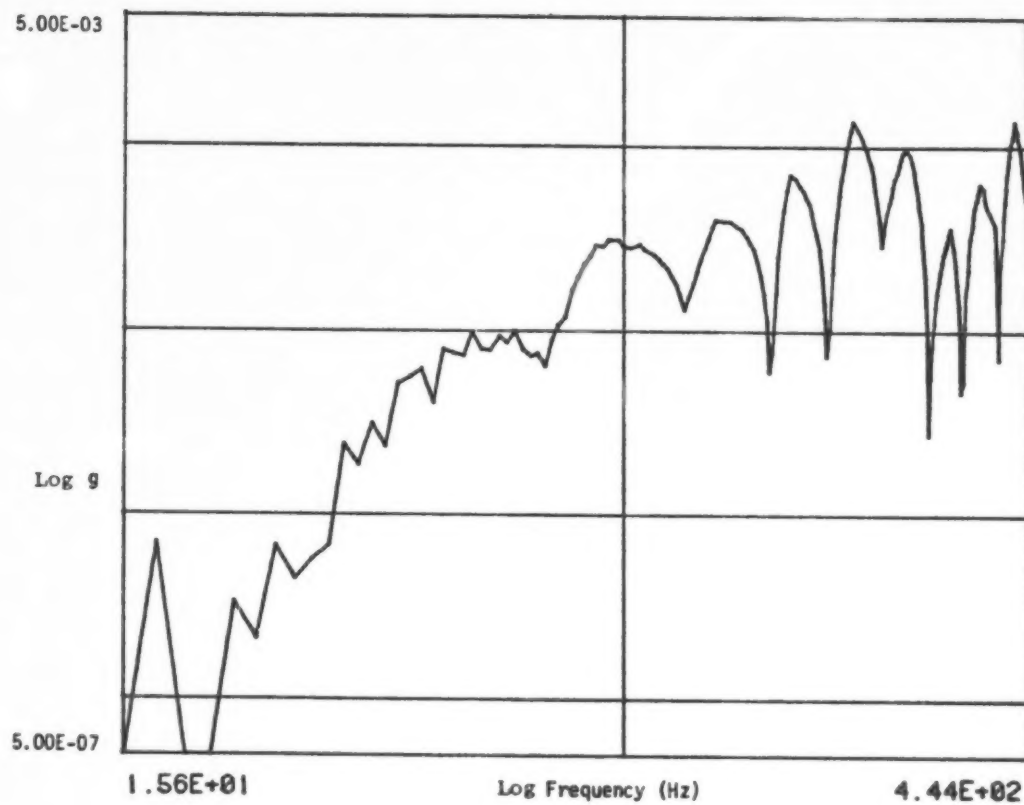


FIGURE 37
VERTICAL VIBRATION RESPONSE OF MODULE TUBE
LOCATION 25

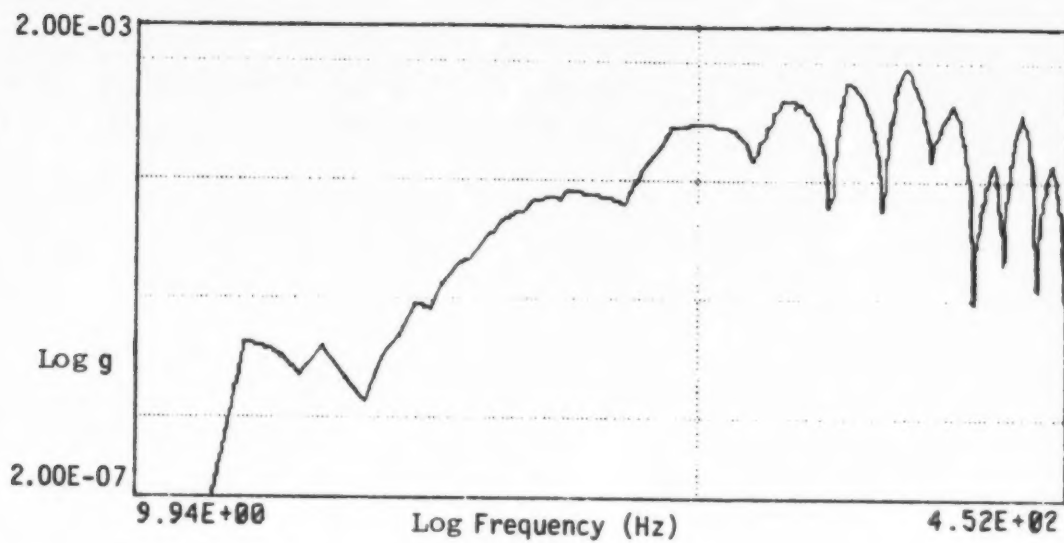


FIGURE 38
VERTICAL VIBRATION RESPONSE OF MODULE TUBE
LOCATION 31

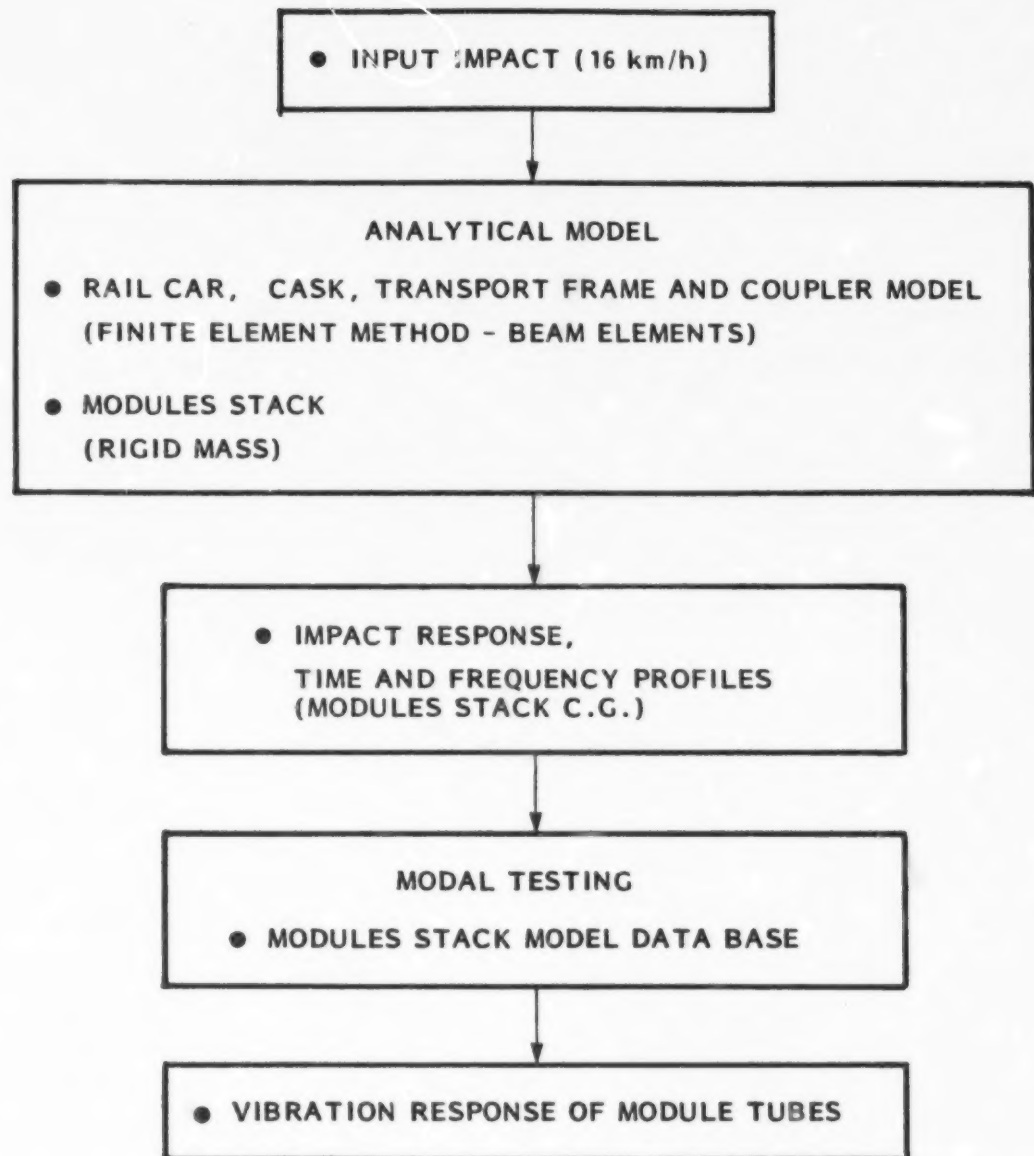


FIGURE 39
MODULE ANALYSIS: RAILCAR END IMPACT MODEL

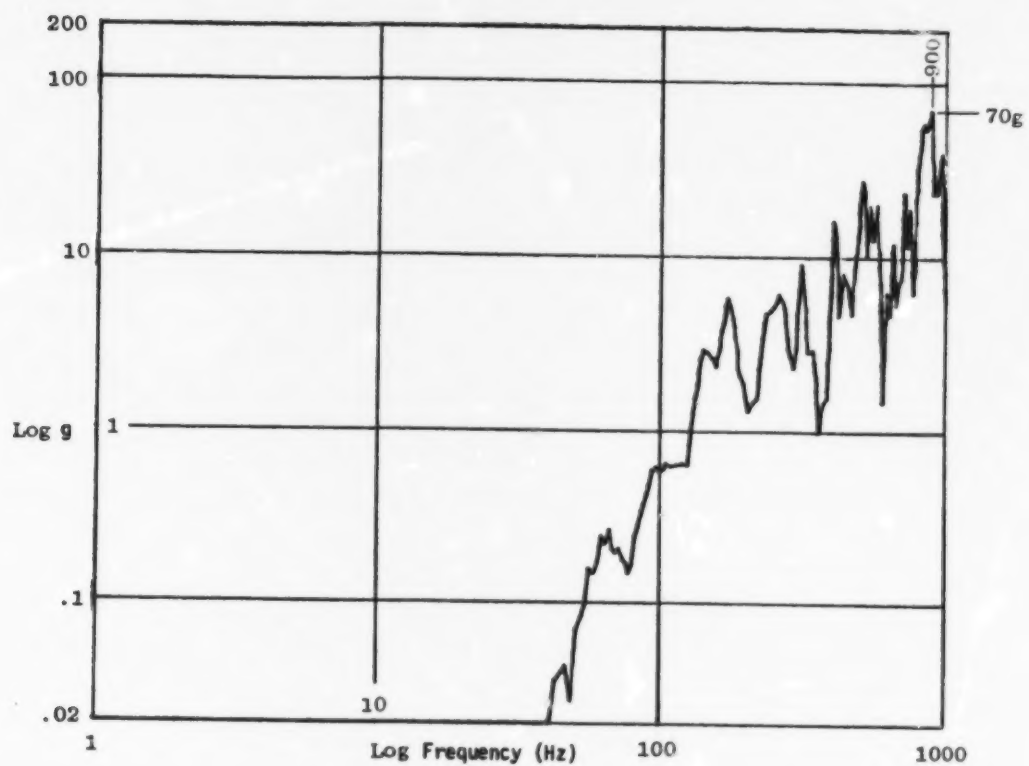


FIGURE 40
VERTICAL SHOCK RESPONSE OF MODULE TUBE
LOCATION 25

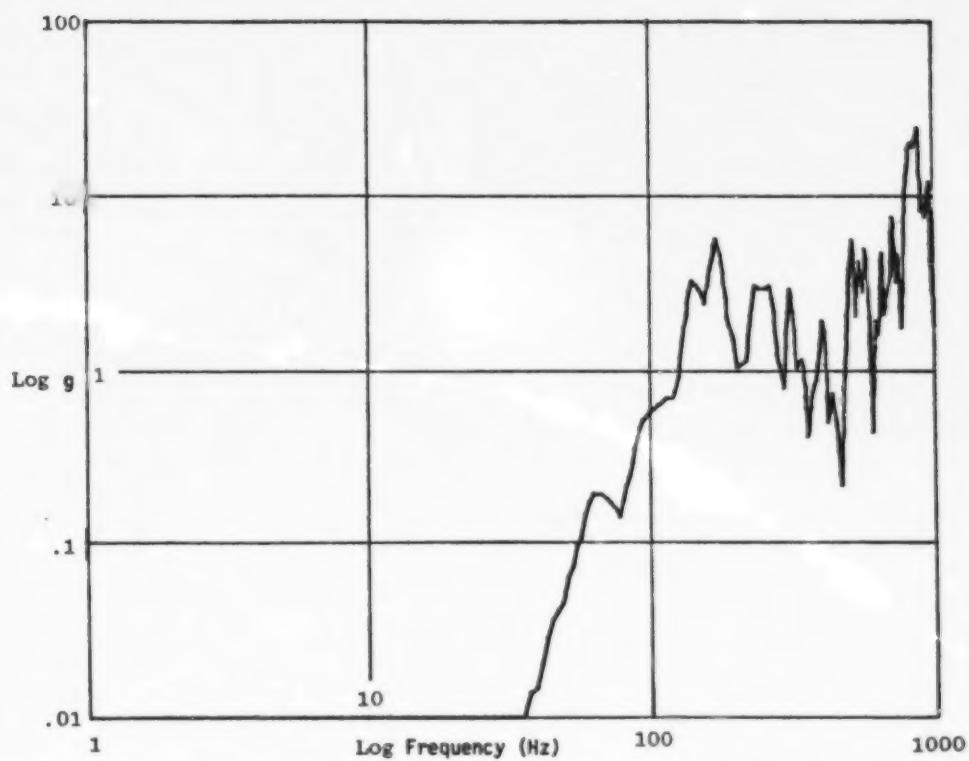
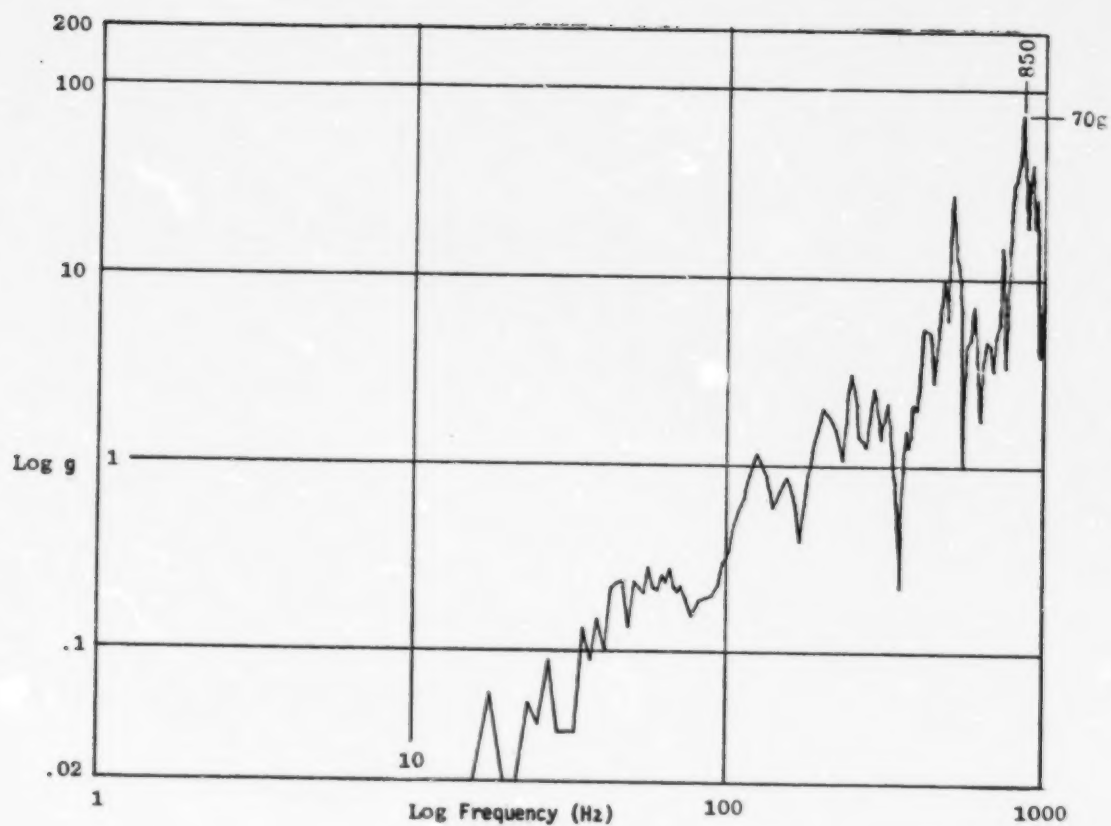


FIGURE 41
VERTICAL SHOCK RESPONSE OF MODULE TUBE
LOCATION 31



LOCATION 25

FIGURE 42
HORIZONTAL SHOCK RESPONSE OF MODULE TUBE

LATERAL TEST FIXTURE

TRANSFER FUNCTION

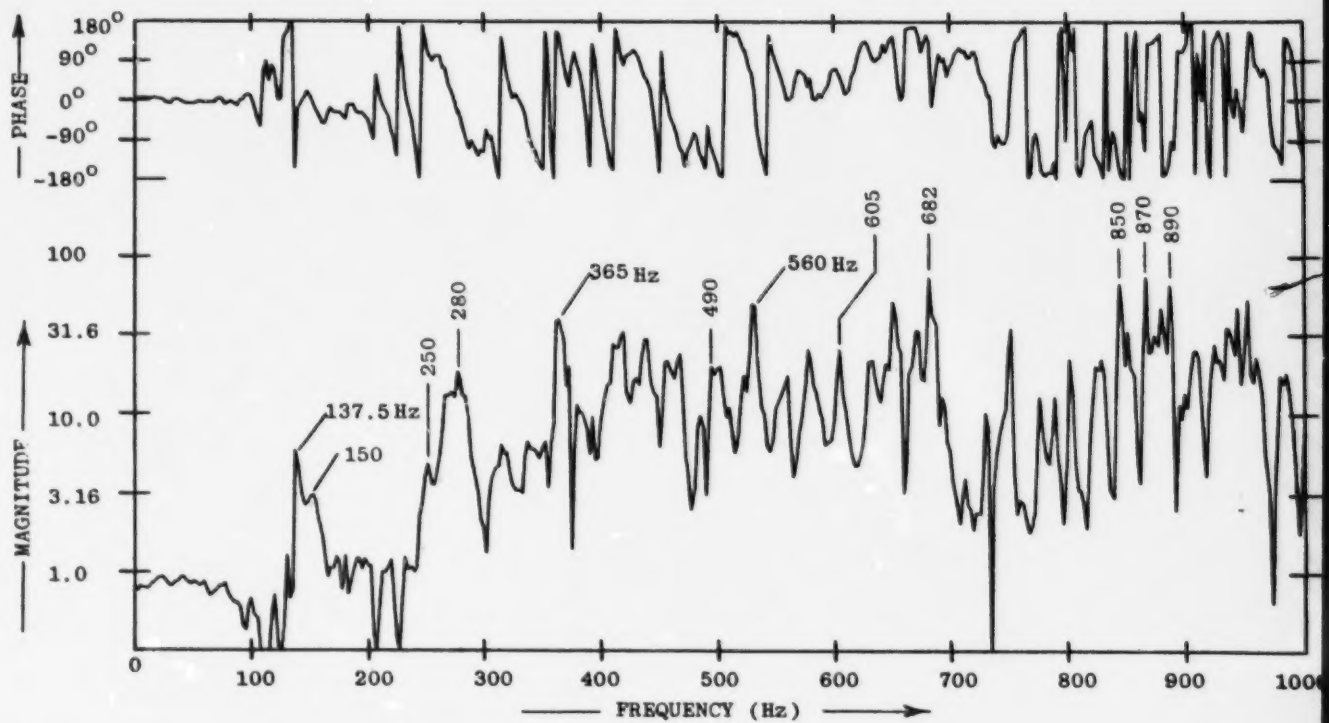
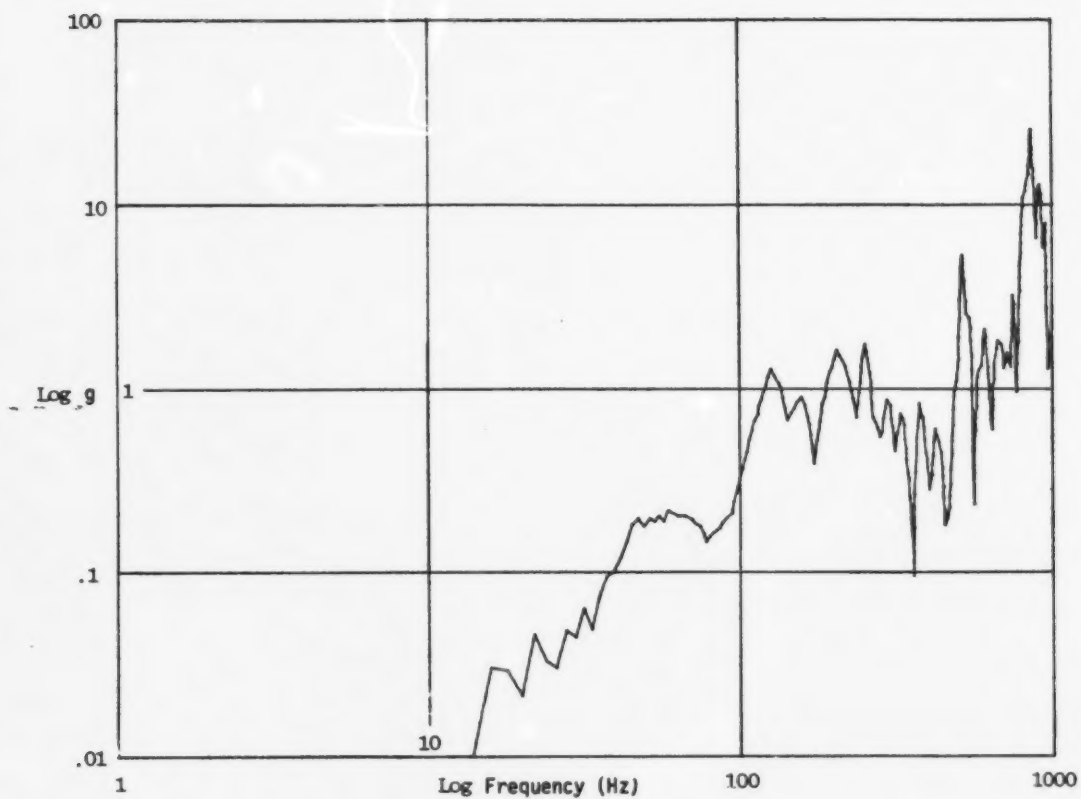


FIGURE 44



LOCATION 31

FIGURE 43
HORIZONTAL SHOCK RESPONSE OF MODULE TUBE